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(NASA-CR-171162) REVIEW OF LOX BEARING AND SEAL MATERIALS TESTER (BSMT) RADIAL LOAD SYSTEM Final Report (Battelle Columbus Labs., Ohio.) 83 p HC A05/MF A01 CSCL 13I

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# Report

FINAL REPORT

on

# REVIEW OF LOX BEARING AND SEAL MATERIALS TESTER (BSMT) RADIAL LOAD SYSTEM
(Contract NAS 8-34908 Task 114)

to

NATIONAL AERONAUTICS AND SPACE
ADMINISTRATION
George C. Marshall Space Flight Center
Marshall Space Flight Center, Alabama

September 28, 1984

by

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September 28, 1984

National Aeronautics and
Space Administration
George C. Marshall Space
Flight Center
Marshall Space Flight Center, AL 35812

Attention: Code AP29-F

Gentlemen:

Contract No. NAS8-34908 (Task No. 114)

Enclosed is the final report on "Review of LOX Bearing and Seal Materials Tester (BSMT) Radial Load System".

Please call me at (614) 424-4618 if there are any questions.

Sincerely,

Keith Dufrane

K. F. Dufrane
Dynamics Section

KFD:llb

Enclosure

xc: See next page for Report Distribution

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FINAL REPORT

on

# REVIEW OF LOX BEARING AND SEAL MATERIALS TESTER (BSMT) RADIAL LOAD SYSTEM

(Contract NAS 8-34908 Task 114)

to

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
George C. Marshall Space Flight Center
Marshall Space Flight Center, Alabama

from

BATTELLE
Columbus Laboratories

September 28, 1984

## INTRODUCTION

Battelle has been assisting NASA-Marshall Space Flight Center (MSFC) on problems concerning the bearings in the high pressure oxygen turbopumps (HPOTP) under a Task Order Agreement. The various tasks have involved failure analyses, bearing dynamics calculations, lubrication studies, wear studies, and analyses of thermal transients. This particular Task involved an analysis of the radial load system on MSFC's bearing and seal tester used to study components for the HPOTP in liquid oxygen (LOX), a brief study of the wear behavior of AISI 440C steel with polytetrafluoroethylene (PTFE) lubrication, and a review meeting of recent bearing developments with NASA, Rocketdyne, and Battelle personnel in attendance.

## SUMMARY

The review of the radial load system, which was conducted by Mechanical Technology, Incorporated, showed the design to be generally adequate and identified a few possible improvements. The AISI 440C wear studies showed that a significant improvement in wear resistance occurs when the surface finish of the sliding members is above 20 μ inches (0.5 μm) $R_A$. Roughnesses of this level are apparently needed to transfer PTFE at a rate sufficient to lubricate the sliding interfaces. The review meeting pointed out the accomplishments and progress to date and identified research areas needed to extend the bearing life closer to the desired design goals.

## REVIEW OF THE LOX BEARING AND SEALS TESTER RADIAL LOAD SYSTEM

The review analysis was conducted by Mechanical Technology, Incorporated (MTI). The report describing the effort is presented in Appendix A. Besides the analysis, MTI personnel visited NASA-MSFC for an initial information-gathering meeting and for a final review meeting.

## LUBRICATED AISI 440C WEAR STUDIES

Wear studies conducted in Task 112 showed that AISI 440C sliding in unlubricated contact against itself wore at rates sufficiently high to explain the ball wear rates occasionally observed in the HPOTP bearings. The current wear tests were intended to explore the effectiveness of PTFE transfer films in controlling the wear rates.

The initial experiments studied the longevity of a PTFE transfer film on a AISI 440C disk used in the three-button experiments. The disk, which had a surface finish of 0.18 μm (7 μ in.), was burnished with PTFE for one hour by substituting three PTFE buttons for the AISI

440C buttons. A total load of 810 N (182 lbs) was applied to the three buttons, and a sliding speed of 0.72 m/s (2.36 ft/s) was used. The button diameter was 7.94 mm (0.3125 in.). The regular AISI 440C buttons were then placed in the holder and the PTFE-coated disk placed in sliding contact with them at a total load of 520 N (117 lbs), a sliding speed of 0.72 m/s (2.36 ft/s), and at room temperature. The continuously recorded friction showed that the PTFE transfer film failed in approximately five minutes. A second experiment with a disk having an identical surface finish failed after three minutes of running. The second disk had the added benefit of a single spring-loaded button of PTFE contacting the disk in the wear track. It was loaded against the disk with a 22 N (5 lb) force provided by a spring. These results showed that a transfer film of PTFE is inadequate to lubricate the AISI 440C at ambient temperatures. Further, a PTFE button could not resupply the surfaces at a sufficient rate.

The influence of surface finish was studied by repeating the above experiments on disks having surface finishes of 0.5 $\mu$m (20 $\mu$in.) and 0.58 $\mu$m (23 $\mu$in.). A surface finish of 0.38 $\mu$m (15 $\mu$ inch) is an established minimum in engineering practice for good transfer of PTFE. The resulting wear lives were increased significantly to 20 minutes and 65 minutes, respectively. Apparently, the initial transfer film was much more durable with the rougher surface finishes. Also, the PTFE button was unable to resupply the track at a sufficient rate to permit indefinite running. The experiments were terminated when the friction began to rise, indicating a lubrication failure. Button wear measurements showed that no measurable wear had occurred, which demonstrates the effectiveness of an adequate PTFE transfer film. For comparison, an identical test run under unlubricated conditions in Task 112 resulted in an average button wear of 1.4 mm (0.055 in.) in 60 minutes.

A graphical representation of the influence of surface finish on PTFE transfer film effectiveness is presented in Figure 1. The sliding distance to failure of the transfer film was converted to equivalent engine running time by assuming an average sliding velocity

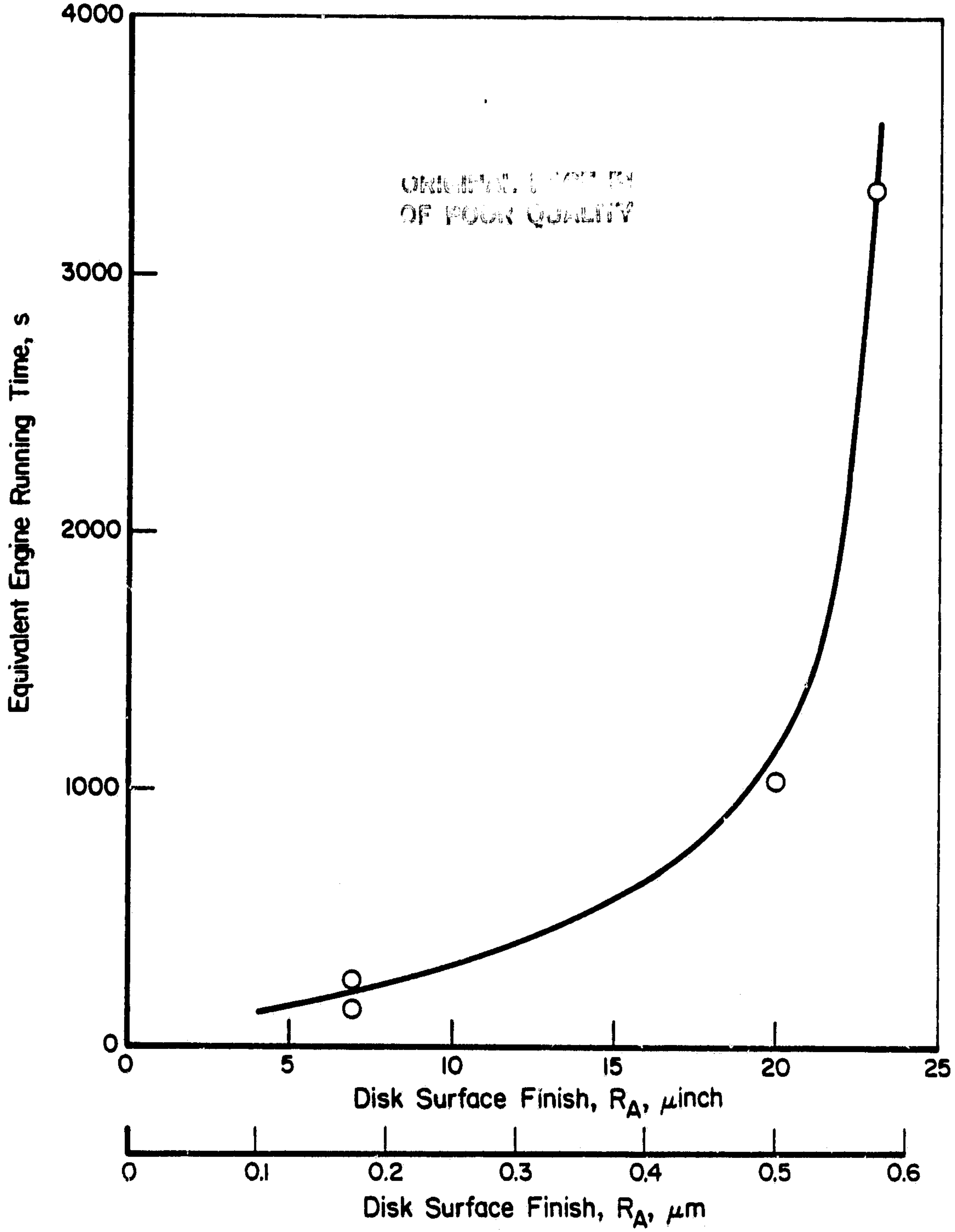


FIGURE 1. IMPROVEMENT OF THE EFFECTIVENESS OF PTFE TRANSFER FILM WITH INCREASED ROUGHNESS OF DISK SURFACE

between the balls and races of 0.81 m/s (32 in./s). Rougher surfaces are clearly associated with extended running times to failure. If a wearing member, such as the cage, is intended to supply the PTFE on a continuing basis, a countering consideration is the wear rate of the supplying member. PTFE wear rates typically increase with increases in roughness of the mating surface. A surface finish of 0.38 μm (15 μ in.) is generally accepted as being a compromise finish that is rough enough to promote transfer without causing excess wear rates on PTFE. These results suggest two important points regarding application to the HPOTP.

1. Lubrication of the bearings may be greatly enhanced by increasing the roughness of the balls and/or races to the range of 0.4 to 0.5 μm (15 to 20 μ in.).

2. If effective PTFE transfer rates can be attained, wear of the element supplying the PTFE (such as the cage) may become substantial and must be considered in the design.

## DISCUSSION OF MEETING BETWEEN NASA/ROCKETDYNE/BATTELLE

The overall objective of the Battelle task order project is to support the NASA efforts to improve the space shuttle main engine (SSME) bearings. Over the course of the last project, five tasks pertaining to this subject have been completed at Battelle which have encompassed three specific bearing-related topics.

- Cage dynamic stability,
- Ball-race lubrication with a cryogenic fluid, and
- Ball wear.

The purpose of the Battelle/NASA/Rocketdyne meeting was to discuss these tasks and the significance of the results with regard to engine performance and improvement.

The general conclusions from the Battelle tasks presented at the meeting were as follows.

- Cage dynamics problems can be controlled by proper selection of ball-cage and cage-race clearances and/or by adequate ball-race lubrication.

- Ball-race lubrication must occur as a result of surface films such as coating or transfer layers from the cage. Any lubrication by cryogenic fluids (LOX, $LN_2$, etc.) is meager, although these fluids do tend to give some surface protection as evidenced by surface evaluation and friction measurement.

- Under the type of sliding conditions seen in a SSME bearing, some level of wear is inevitable. This wear can easily be on the order of $5 \times 10^{-6}$ inch/second, which will greatly limit bearing life.

Battelle's view, then, is that the key to adequate bearing performance is good transfer film lubrication. Wear can be virtually eliminated if a transfer film can be generated in the ball wear track. Very little success with transfer films has been achieved with smooth race or ball surfaces. However, as discussed previously,, surfaces with rough finishes (~15 $\mu$inch) have generated good transfer films.

There are several activities being pursued by Rocketdyne/NASA to improve bearing performance which include:

- Ion implantations,
- Improved surface coatings on the bearing races and/or balls (titanium nitride and cubic boron nitride),
- Improved bearing steels (one possibility is M-50 steel with a corrosion-resistant film), and
- Reduction of preload to 485 lbs axial.

The SSME bearings can currently operate at 104 percent capacity for three to four flights (550 seconds/flight) and each new engine endures 1300 to 2000 seconds of testing prior to flight. Therefore, the bearings can operate with current technology for a total 2000 to 4000 seconds at 104 percent. The desired target is 27,000 seconds at 109 percent.

Our observations are that important work is being done in improving the bearing materials and initial coatings. These efforts will doubtless result in improved bearing performance. However, the level of improvement required to achieve the 27,000 seconds at 109 percent capacity will require much more work in transfer film lubrication. Future tasks at Battelle directed at this subject could greatly supplement the NASA/Rocketdyne activities. Specific tasks could include:

- Evaluate the specific level of surface roughness on balls or races required to promote solid lubricant transfer.
- Evaluate effect of surface finish on cage wear.
- Survey people involved in transfer film studies to ascertain that we are using the best available information.
- Evaluate alternate design concepts for lubricant transfer (c.f. a design where the cage rubs on the ball track surface of the race).
- Evaluate suitability of other candidate solid lubricant coatings.

## MEASURING AND CALCULATING UNITS

Since the bearing drawings and all input data provided by NASA were in English units, all measurements and calculations were performed in English units. The SI units represented in this report were converted from English units. The data on which the report is based are located in Battelle Laboratory Record Book 39381.

# APPENDIX A

## REVIEW OF THE LOX BEARING AND SEALS MATERIAL TESTER (BSMT) RADIAL LOAD SYSTEM, BY MECHANICAL TECHNOLOGY INCORPORATED

MTI 84TR65

# REVIEW OF LOX BEARING AND SEALS
# MATERIAL TESTER (BSMT)
# RADIAL LOAD SYSTEM

July 1984

Prepared by:
W. Shapiro
Senior Staff Consultant

Prepared for:
Battelle, Columbus/NASA/MSFC

MECHANICAL TECHNOLOGY INCORPORATED
968 Albany-Shaker Road
Latham, New York 12110

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## 1.0 INTRODUCTION

Mechanical Technology Incorporated (MTI) has completed the review of the LOX Bearing and Seal Materials Tester (BSMT) Radial Load System designed by NASA/MSFC. The objectives were to: (1) review the NASA-designed LOX BSMT Hydraulic Bearing Radial Load Concept as shown on drawing #30A85200-1 and details and (2) provide a written assessment concerning the feasibility of the design with recommendations for improvement as required to meet the operating constraints, fluids used, and the loading requirements as identified in BSMT-URD-83-3, dated March 25, 1983. Special attention to safety of operation in LOX, ability to achieved desired loading, flow requirements, instrumentation recommendations and an assessment of the load predictability was desired. Requirements were identified in BSMT-URD-82-3. The complete work statement is included in Appendix B. Pertinent information was delivered to MTI representatives at a meeting held at NASA/MSFC on June 13, 1984. A presentation of MTI's findings was made on July 19, 1984 at NASA/MSFC. This report is basically a compilation of the presentation material with some elaboration.

## 2.0 SUMMARY OF RESULTS AND CONCLUSIONS

The hydrostatic radial load device should operate satisfactorily with some possible reservations noted directly below:

- Pressure probes and surface grid holes can trap gas before liquefaction takes place. Pneumatic hammer may occur. If this does happen, it will be necessary to prevent the shaft from vibrating until all fluid has liquified.
- Pressure probes will have significant amount of vapor in instrumentation lines which can offset symmetrical equilibrium and load the shaft to one side.

- Load is affected by clearance, eccentricity, and shaft bending. Pressure symmetry is also reduced by bending coupled with rotation. Integrated loads can be in error by approximately 10%.

- Anticipate large flows and potential fluctuations. Should have 70-80 GPM (10.4 $\rightarrow$ 11.9 #/S) Capacity.

- Variation of load with clearance $\geq$ 5%.

  Variation of Flow with clearance $\geq$ 86%

  Variation of Load with eccentricity, $\varepsilon$ (0.4) = - 1.14%

  Variation of Flow with eccentricity, $\varepsilon$ (0.4) = - 34%

  Effect of shaft bending:

  Flow increase = 69%
  Load increase = 3.8%
  Load offset $\sim$ 4 degrees

- Incorporation of a restrictor element near the loading recess will add stiffness and damping (no computations made). Also constant flow in lieu of constant pressure will significantly increase the stiffness and damping of the loading device and is a recommended course of action, as it will provide significantly greater insurance against shaft contact.

- Approximate analysis accomplished independently of the more detailed computer studies indicated that the diamond grid pattern would achieve the highest load and lowest flow.

- Negative stiffness is predicted for the present configuration, but is more than offset by positive damping. Negative stiffness will not occur with constant flow rather than constant pressure.

- Cursory rotor dynamic studies indicate agreement with MSFC critical speeds and mode shapes.

# 3.0 HYDROSTATIC LOADER

## 3.1 HYDROSTATIC LOADER THEORY

- BASIC COMPUTER CODE DESCRIBED IN REFERENCE BELOW:

**ANALYSIS OF HYBRID, FLUID-FILM JOURNAL BEARINGS WITH TURBULENCE AND INERTIA EFFECTS**

by

A. Artiles
W. Shapiro
J. Walowit
(Mechanical Technology Incorporated)

Published in

Advances in Computer-Aided Bearing Design
(ASME Publication)

### ABSTRACT

A method of analyzing performance of hybrid (combined hydrodynamic-hydrostatic) journal bearings, including turbulence in the film and inertia at recess boundaries, is described. The Newton-Raphson method of iteration was found to converge efficiently when treating multiple recesses, high eccentricity ratios, and effects of inertia, misalignment, turbulence, and cavitation (either separate or in combination). The column matrix method applied to a variable-size finite difference grid is used to solve the governing lubrication equations at interior field points. The method and logic are described and several sample problems directed towards cryogenic applications, demonstrating the effectiveness of the approach are presented.

# REYNOLDS EQUATION

$$\frac{\partial}{\partial \theta}\left(H^3 G_x \frac{\partial P}{\partial \theta}\right) + \frac{\partial}{\partial Z}\left(H^3 G_z \frac{\partial P}{\partial Z}\right) = \Lambda \frac{\partial H}{\partial \theta} + \frac{\partial H}{\partial T}$$

$H = h/c$ $\quad$ $h$ = local film thickness

$c$ = clearance

$P = p/p_o$ $\quad$ $p$ = local pressure

$p_o$ = reference pressure

$Z = z/R$ = dimensionless axial coordinate

$\Lambda = 6\mu\omega R^2/(C^2 p_o)$ $\quad$ $\mu$ = viscosity

$\omega$ = rotational speed, r/s

$T = t/(12\mu R^2/C^2 p_o)$ = dimensionless time

$G_x$ = turbulent coefficient, $\theta$ direction

$G_z$ = turbulent coefficient, z direction

- General equation solved by computer code.
- G factors modified for roughness effects.

G's Modified with Child's

Friction Factors*

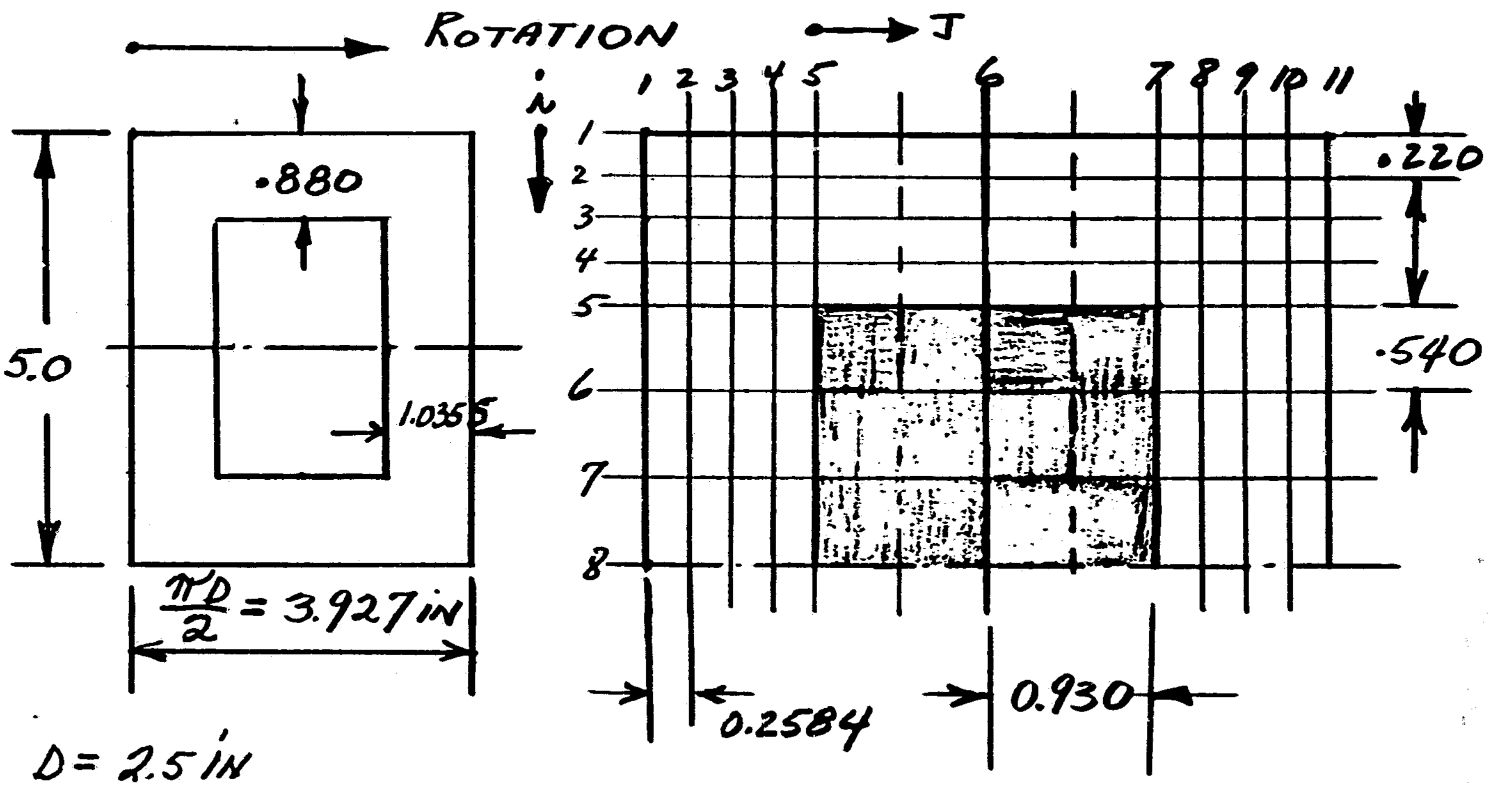
| | Empirical Coefficients | | Relative Roughness |
|---|---|---|---|
| | ms | ns | $\varepsilon = e/2C$ |
| Smooth $C_r$ = .394 mm | -.21663 | .06736 | .0003866 |
| Smooth $C_r$ = .527 mm | -.23980 | .09885 | .0006919 |
| Rocketdyne $C_r$ = .527 mm | -.13567 | .06968 | .022 |
| Diamond-Grid $C_r$ = .527 mm | -.03498 | .11815 | .45973 |
| Hole Pattern $C_r$ = .527 mm | .01904 | .015027 | .05752 |

Table 1. Empirical turbulence coefficients ms, ns for damper seal configurations, and estimates for relative roughness from Colebrook's formula [9] at $R_a$ = 300,000.

Derivation by J. Walowit Appended

*SSME Interstage Seal Research Progress Report, Contract NAS8-33716, January 1984, RD-1-84 - Texas A & M, Turbomachinery Lab, ME Dept.

# HYDROSTATIC LOADER COMPUTER MODEL




$D = 2.5$ IN

$L = 5.0$ IN

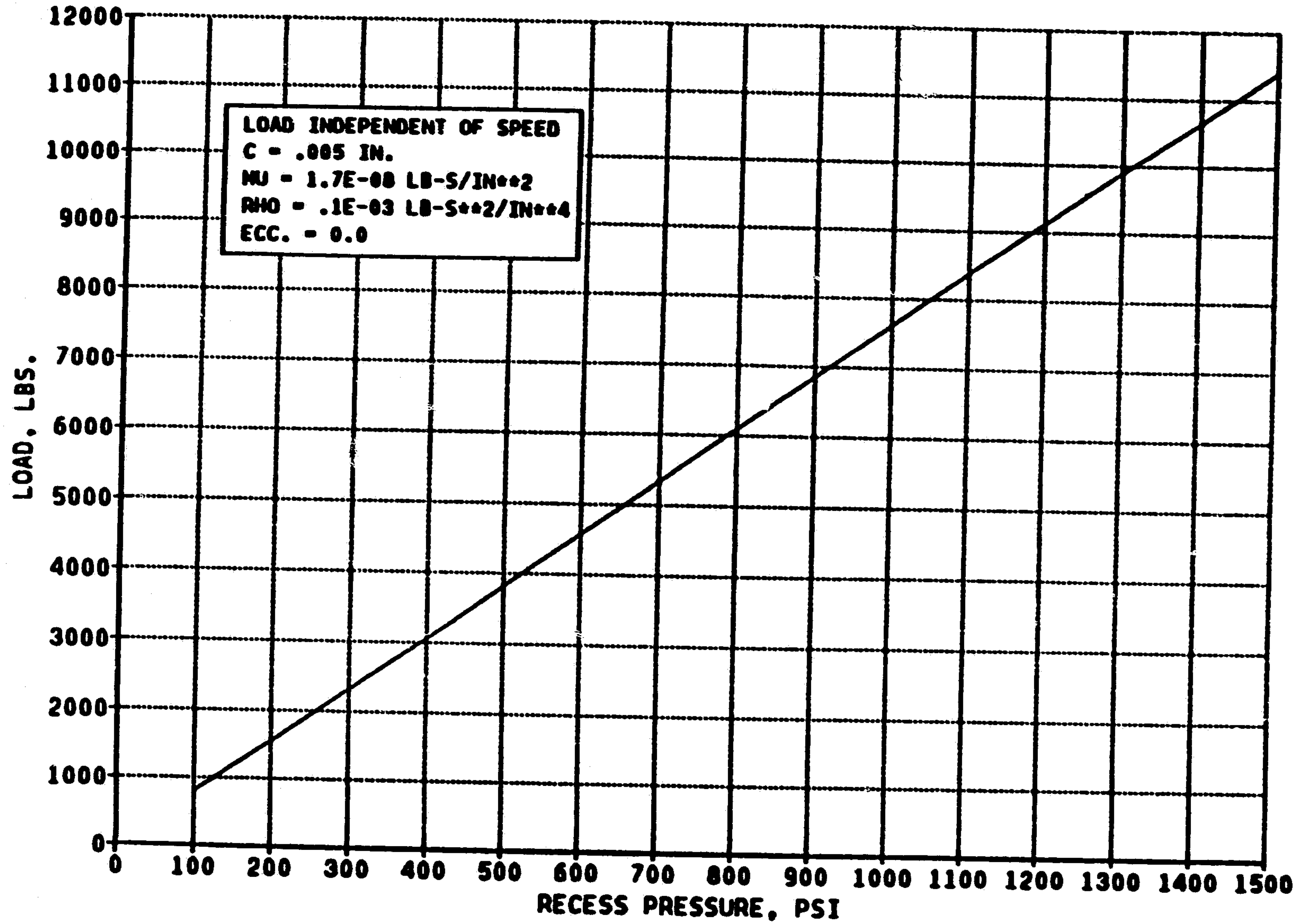
- In most cases 11 x 8 symmetrical grid used.
- In some cases 13 x 8 symmetrical grid used.

## 3.2 PERFORMANCE AS A FUNCTION OF SPEED AND PRESSURE

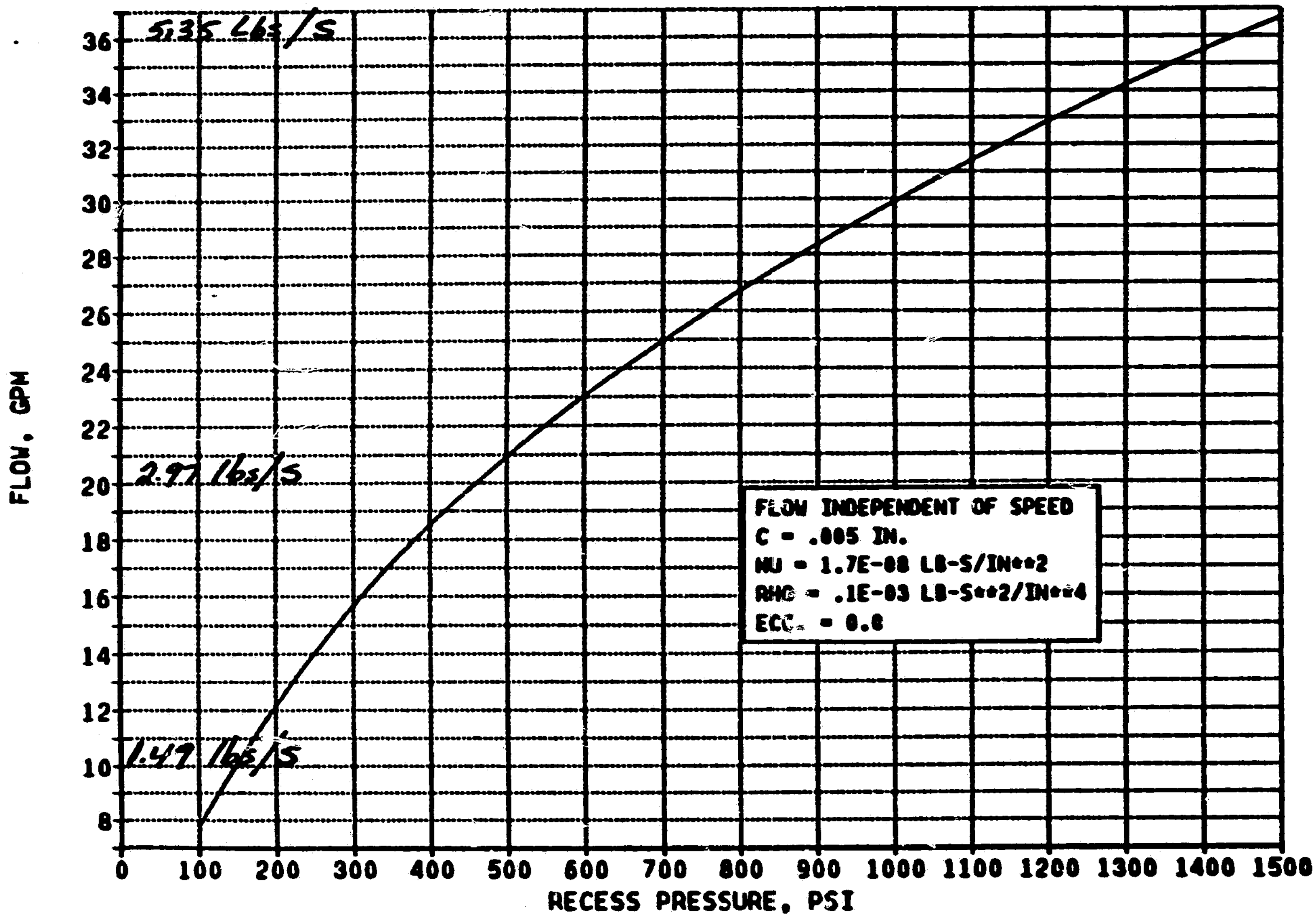
$C = 0.005$

$\varepsilon = 0.0$

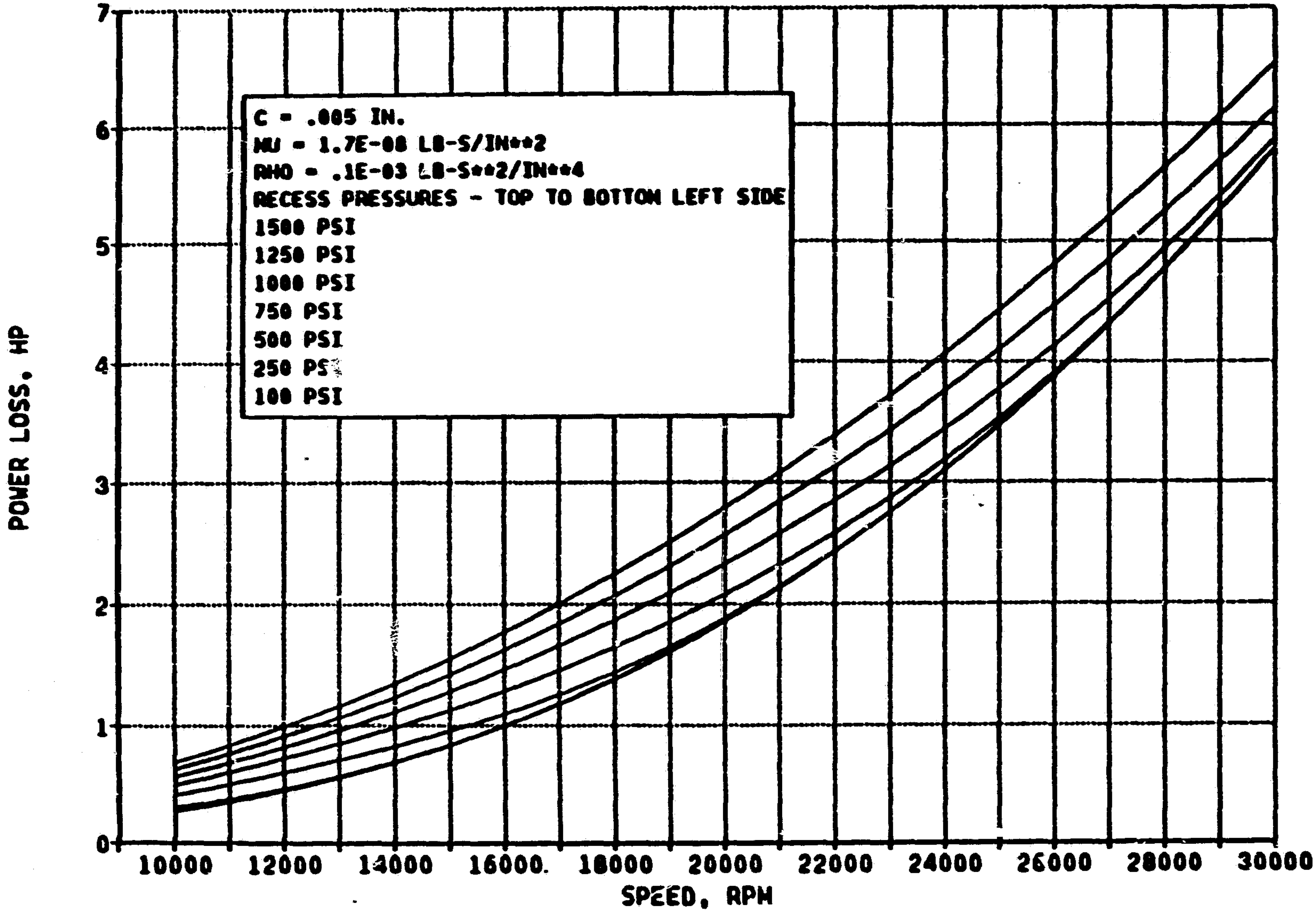
HYDROSTATIC LOADER, LOAD VS. PRESSURE



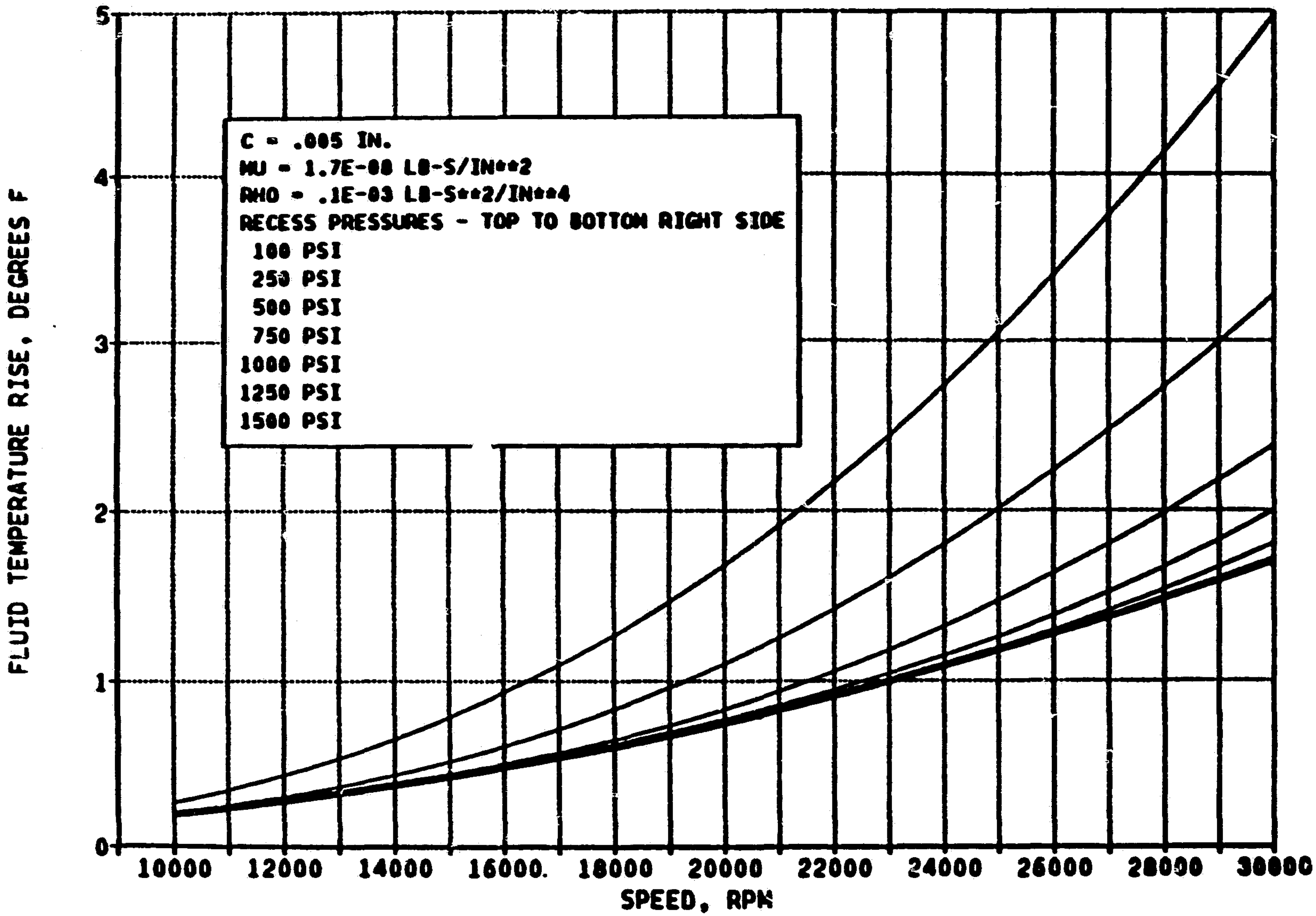
HYDROSTATIC LOADER, FLOW VS. PRESSURE.



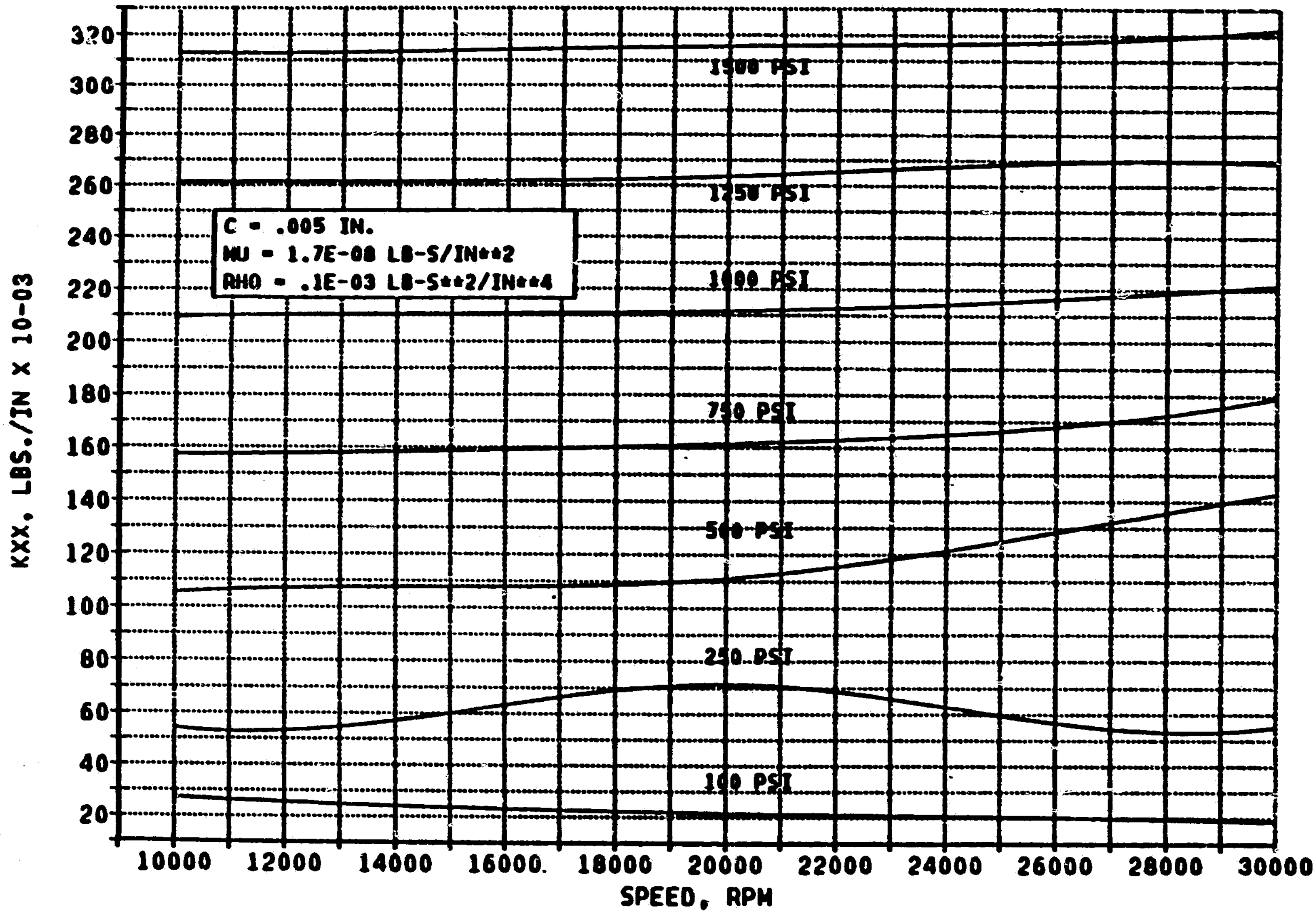
HYDROSTATIC LOADER, POWER LOSS VS. SPEED



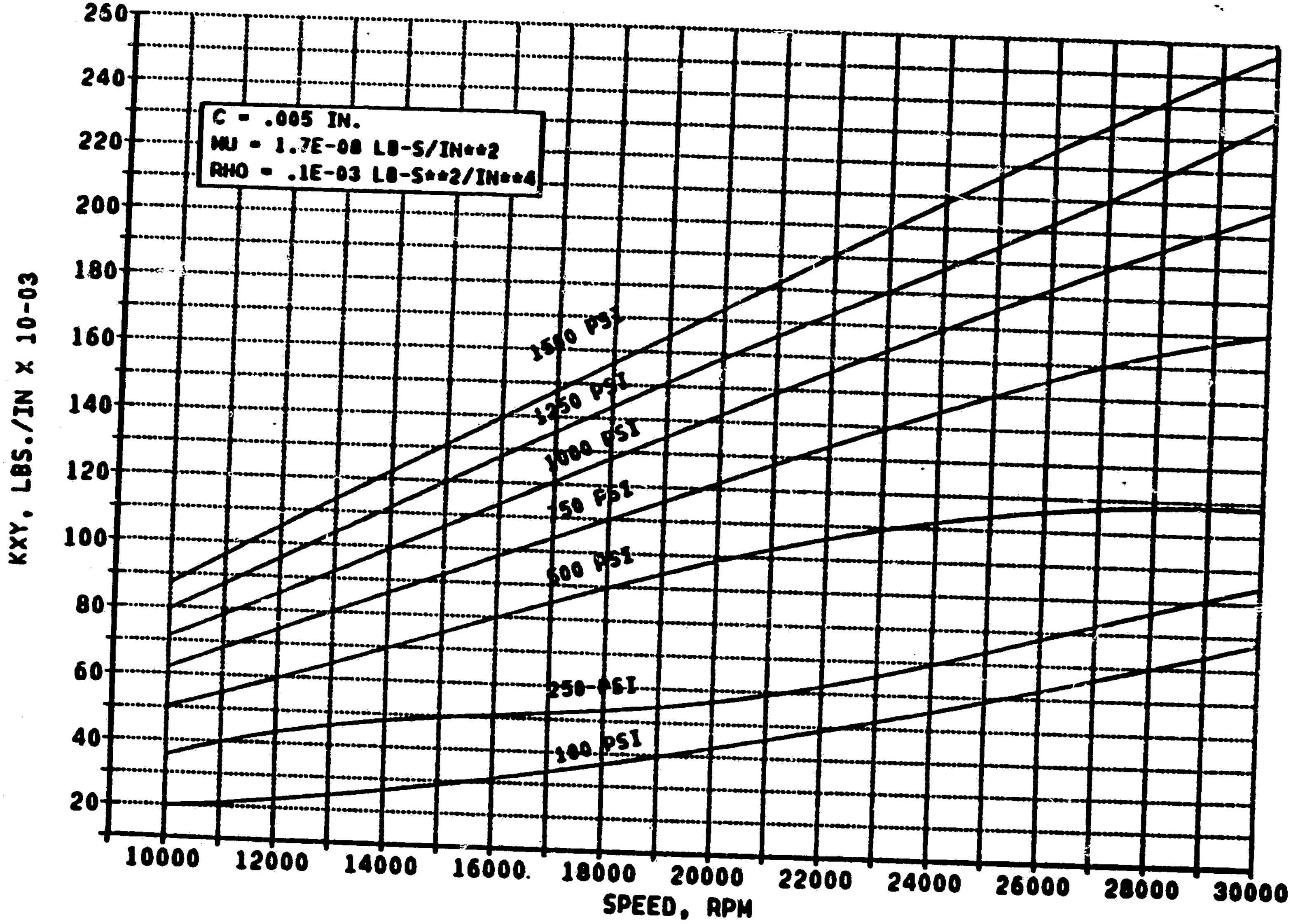
HYDROSTATIC LOADER, FLUID TEMPERATURE RISE VS. SPEED



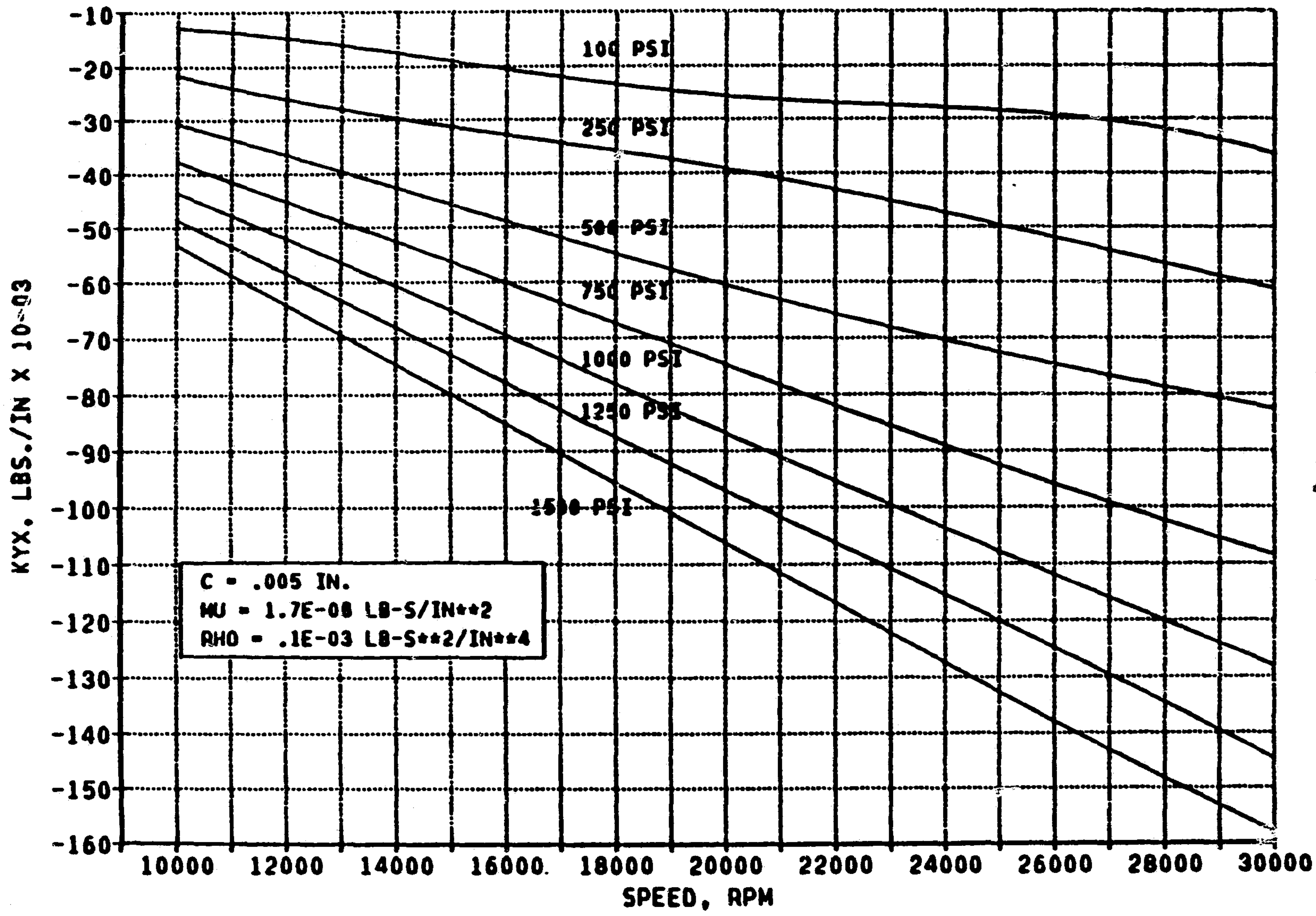
## HYDROSTATIC LOADER, KXX VS. SPEED



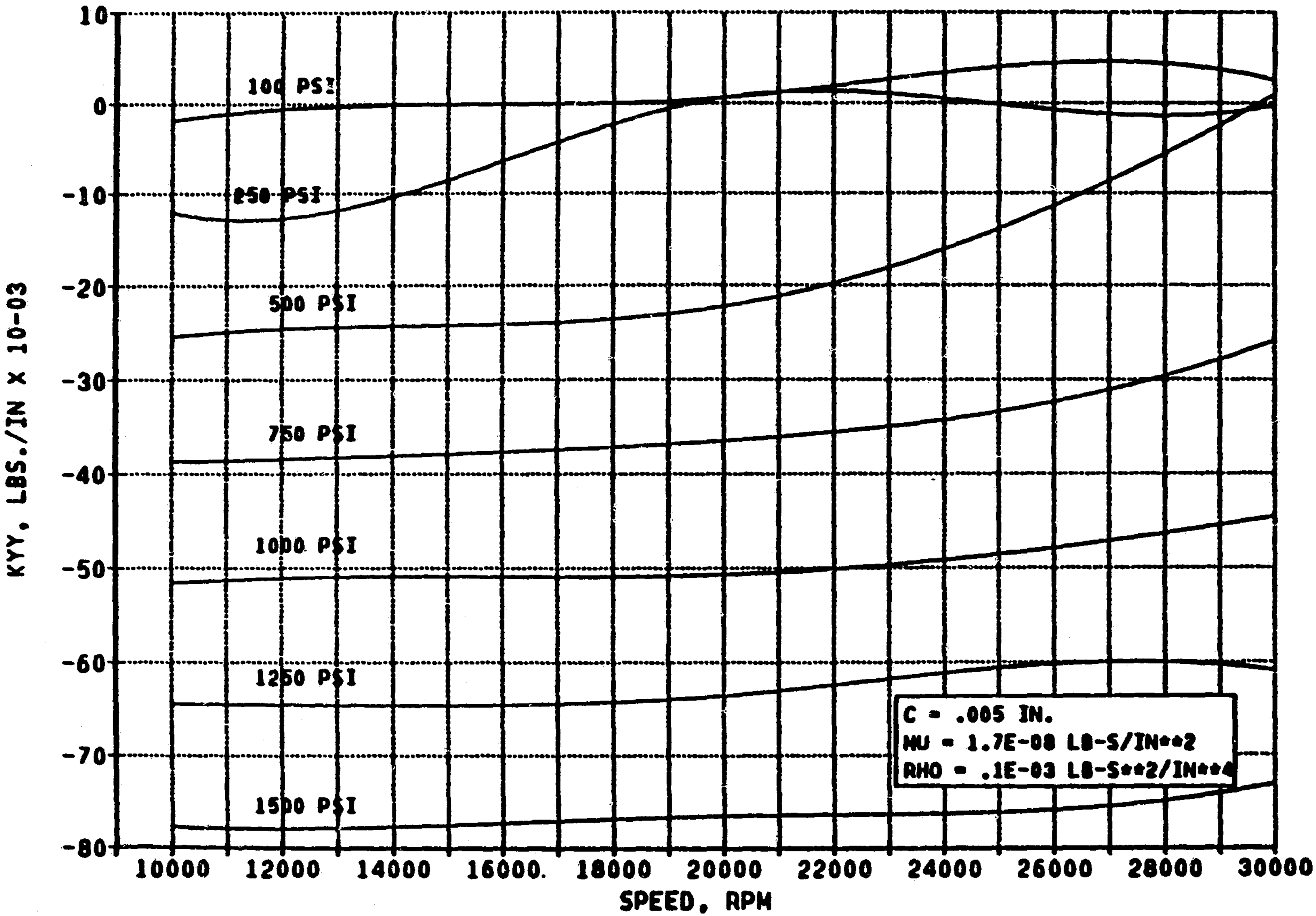
HYDROSTATIC LOADER, KXY VS. SPEED



HYDROSTATIC LOADER, KYX VS. SPEED



# HYDROSTATIC LOADER, KYY VS. SPEED

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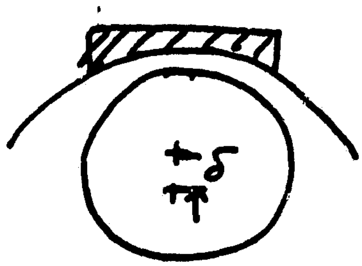
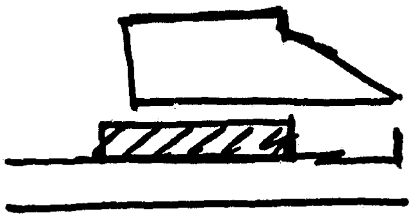
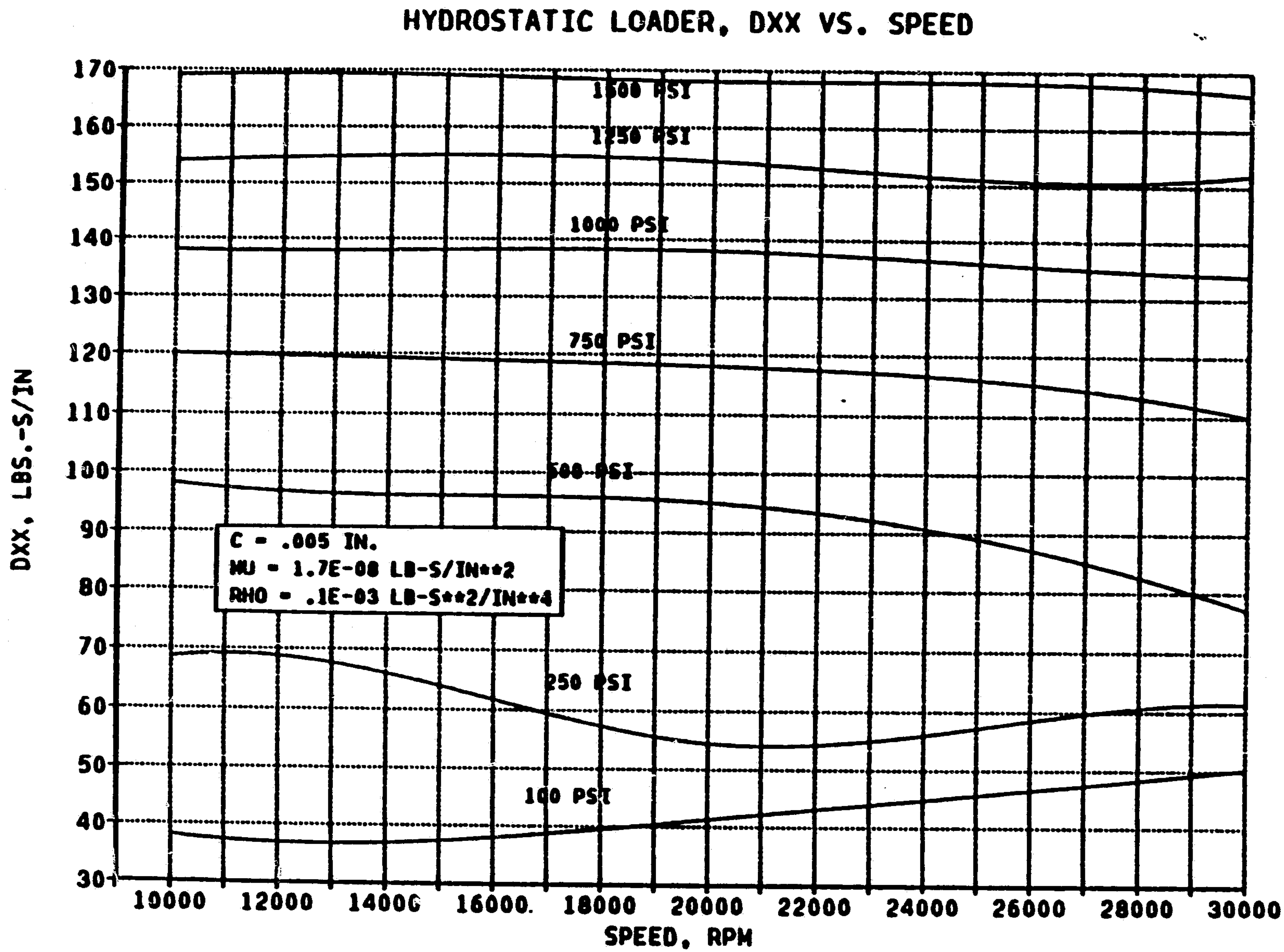


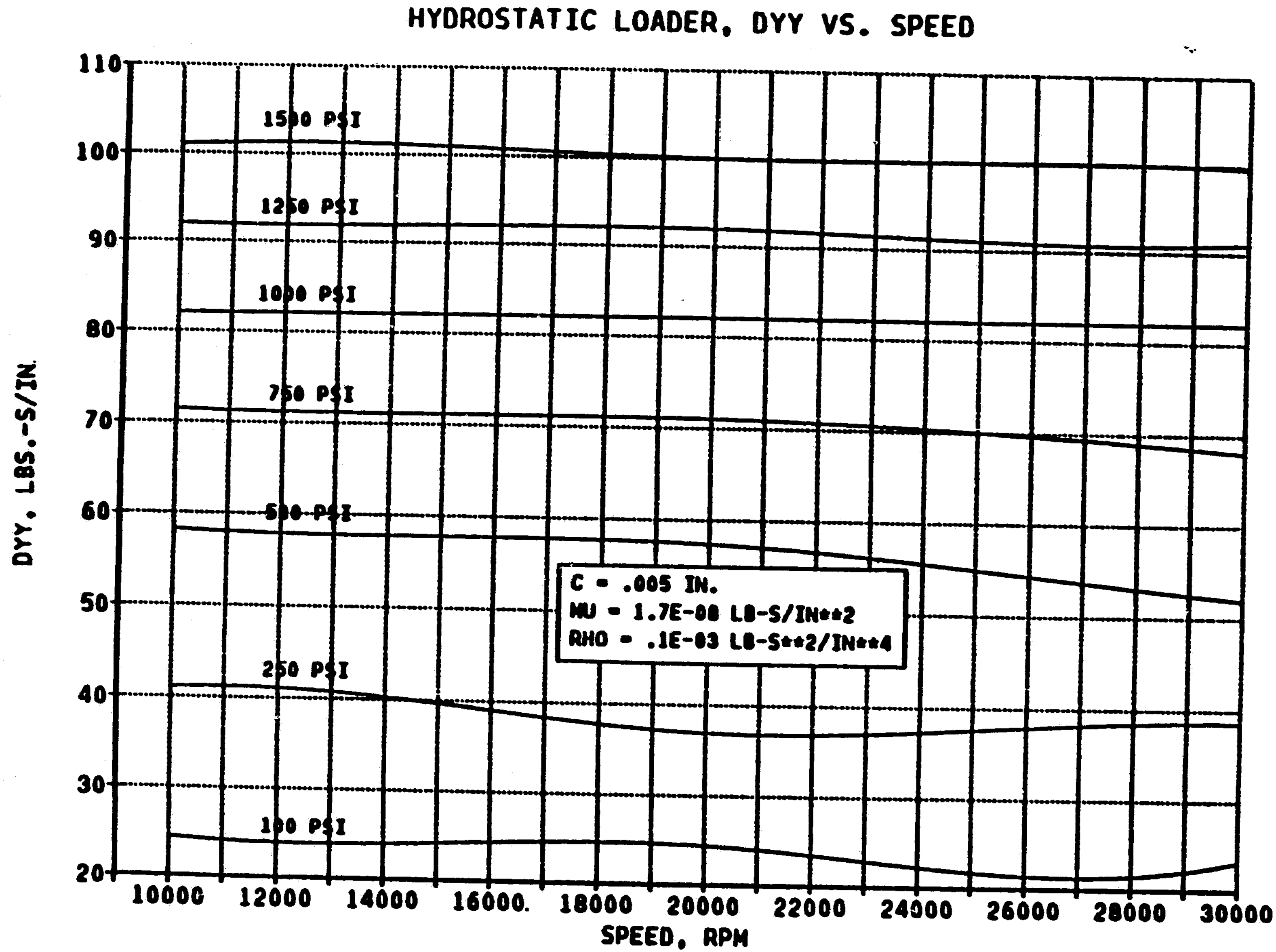

ECCENTRICITY CAN PRODUCE NEGATIVE STIFFNESS. DIVERGENT FILM OVER LANDS.



INERTIA -
INERTIA DROP OVER LANDS ARE LESS WITH SMALLER FILMS
- POSITIVE STIFFNESS

ROUND HOLES PRODUCE OVERALL NEGATIVE STIFFNESS - OTHER CONFIGURATIONS DO NOT.


HYDROSTATIC LOADER, DXX VS. SPEED
1500 PSI
1250 PSI
1000 PSI
750 PSI
500 PSI
250 PSI
100 PSI
C = .005 IN.
MU = 1.7E-08 LB-S/IN**2
RHO = .1E-03 LB-S**2/IN**4
DXX, LBS.-S/IN
170
160
150
140
130
120
110
100
90
80
70
60
50
40
30
10000
12000
14000
16000
18000
20000
22000
24000
26000
28000
30000
SPEED, RPM

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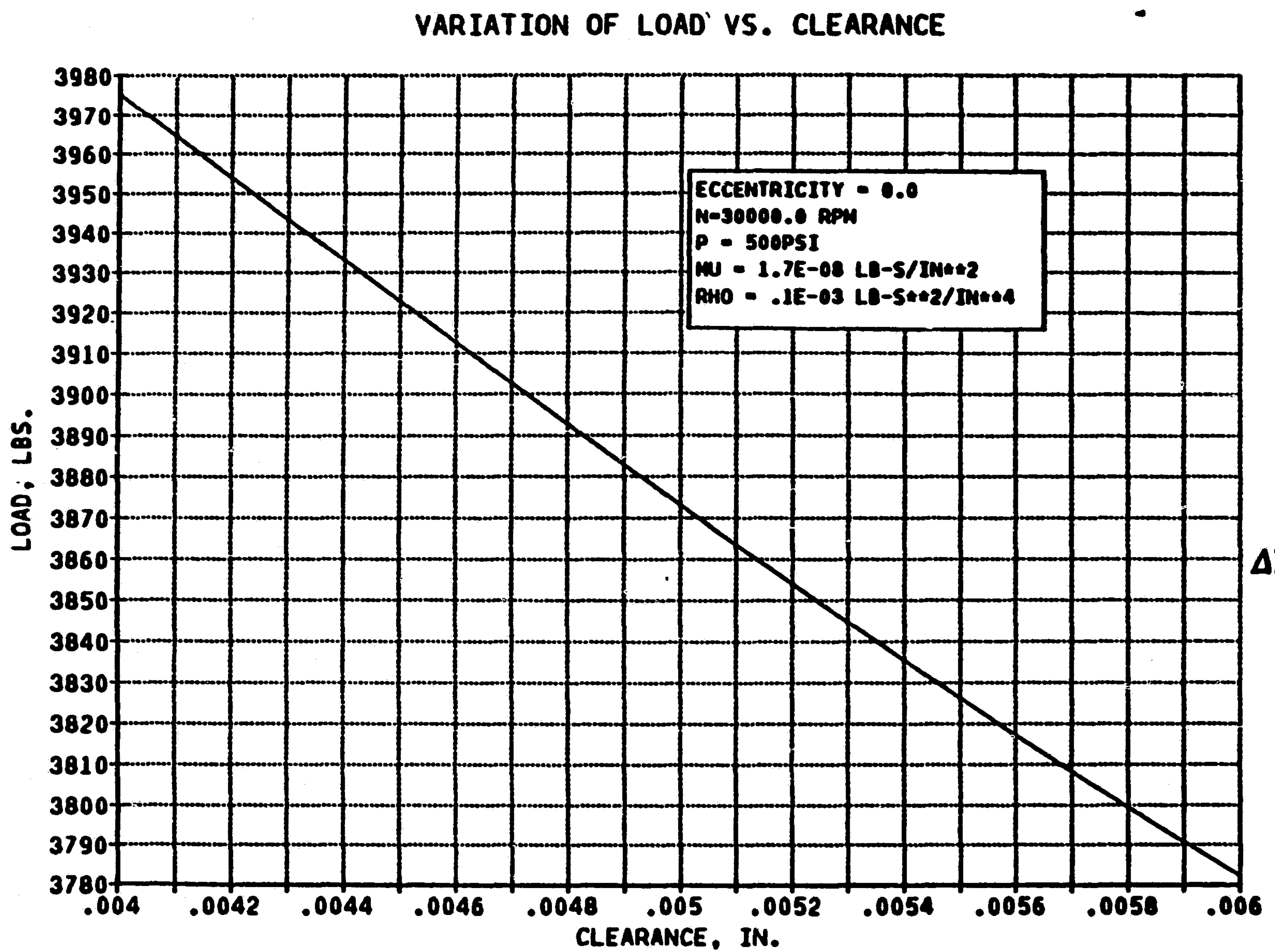
## 3.3 VARIATION OF PERFORMANCE WITH CLEARANCE

$\varepsilon = 0.0$

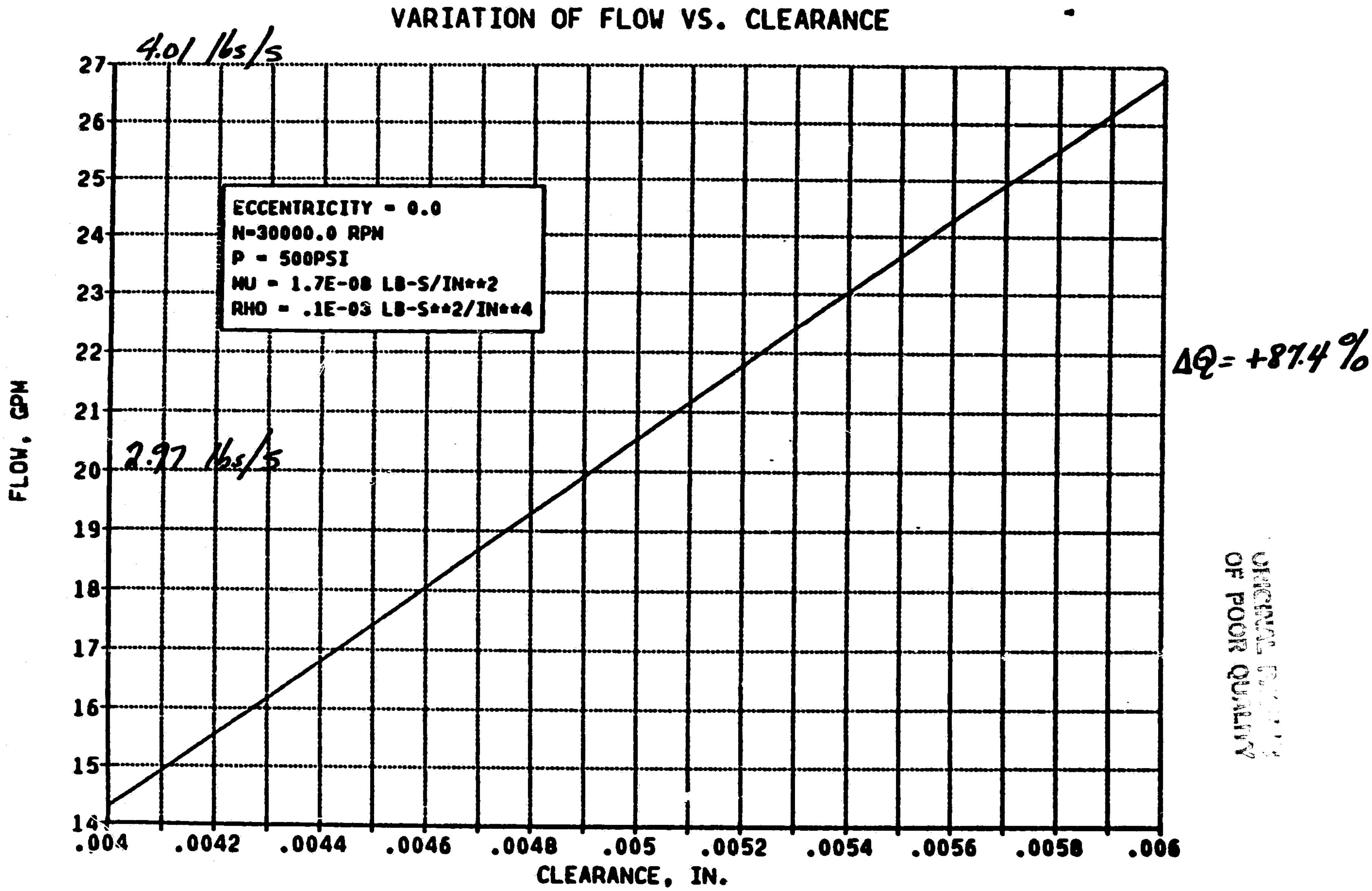
$P = 500$ PSI

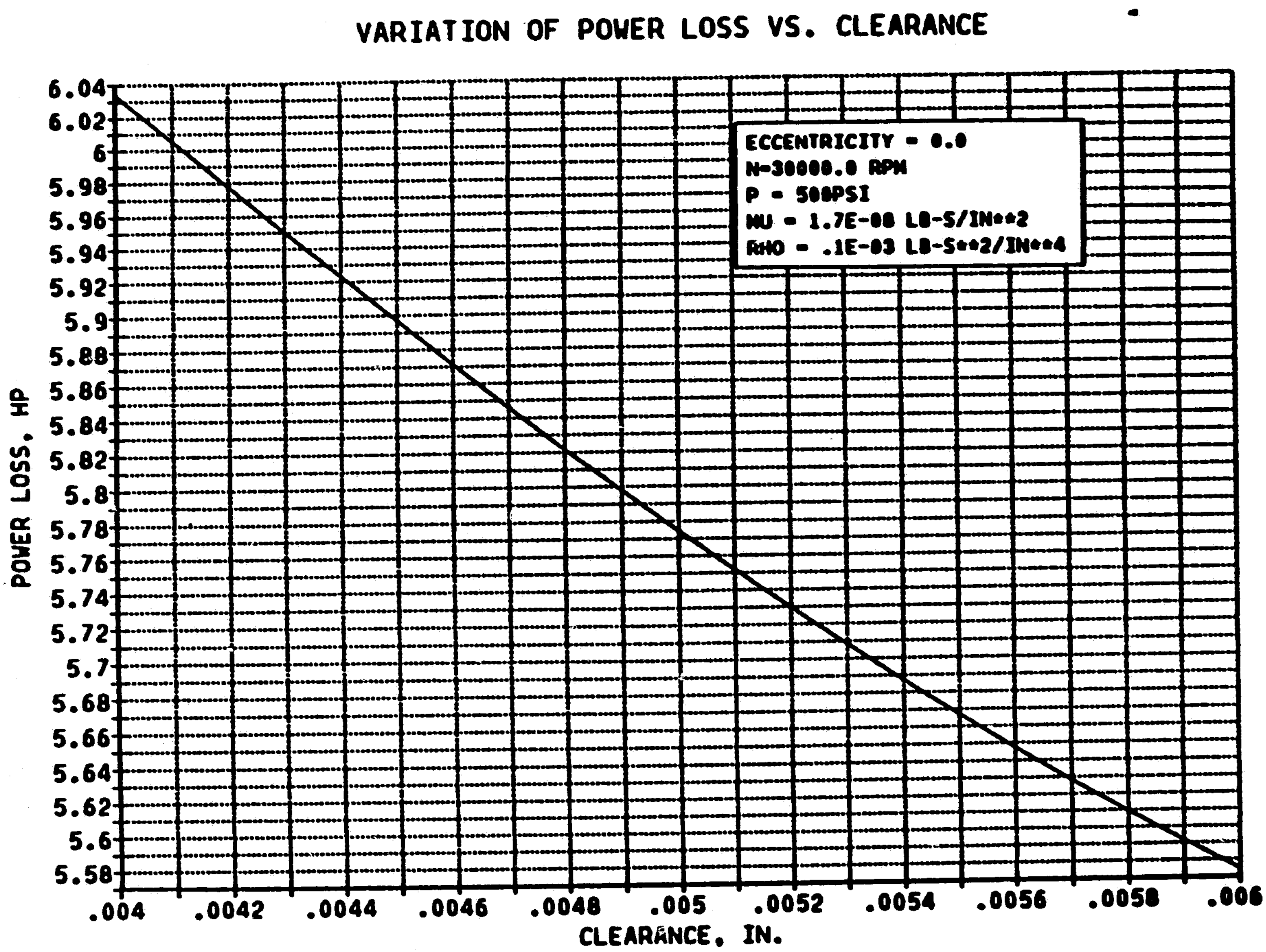
$N = 30{,}000$ RPM



VARIATION OF LOAD VS. CLEARANCE
ECCENTRICITY = 0.0
N=30000.0 RPM
P = 500PSI
MU = 1.7E-08 LB-S/IN**2
RHO = .1E-03 LB-S**2/IN**4
LOAD, LBS.
3980
3970
3960
3950
3940
3930
3920
3910
3900
3890
3880
3870
3860
3850
3840
3830
3820
3810
3800
3790
3780
.004
.0042
.0044
.0046
.0048
.005
.0052
.0054
.0056
.0058
.006
CLEARANCE, IN.

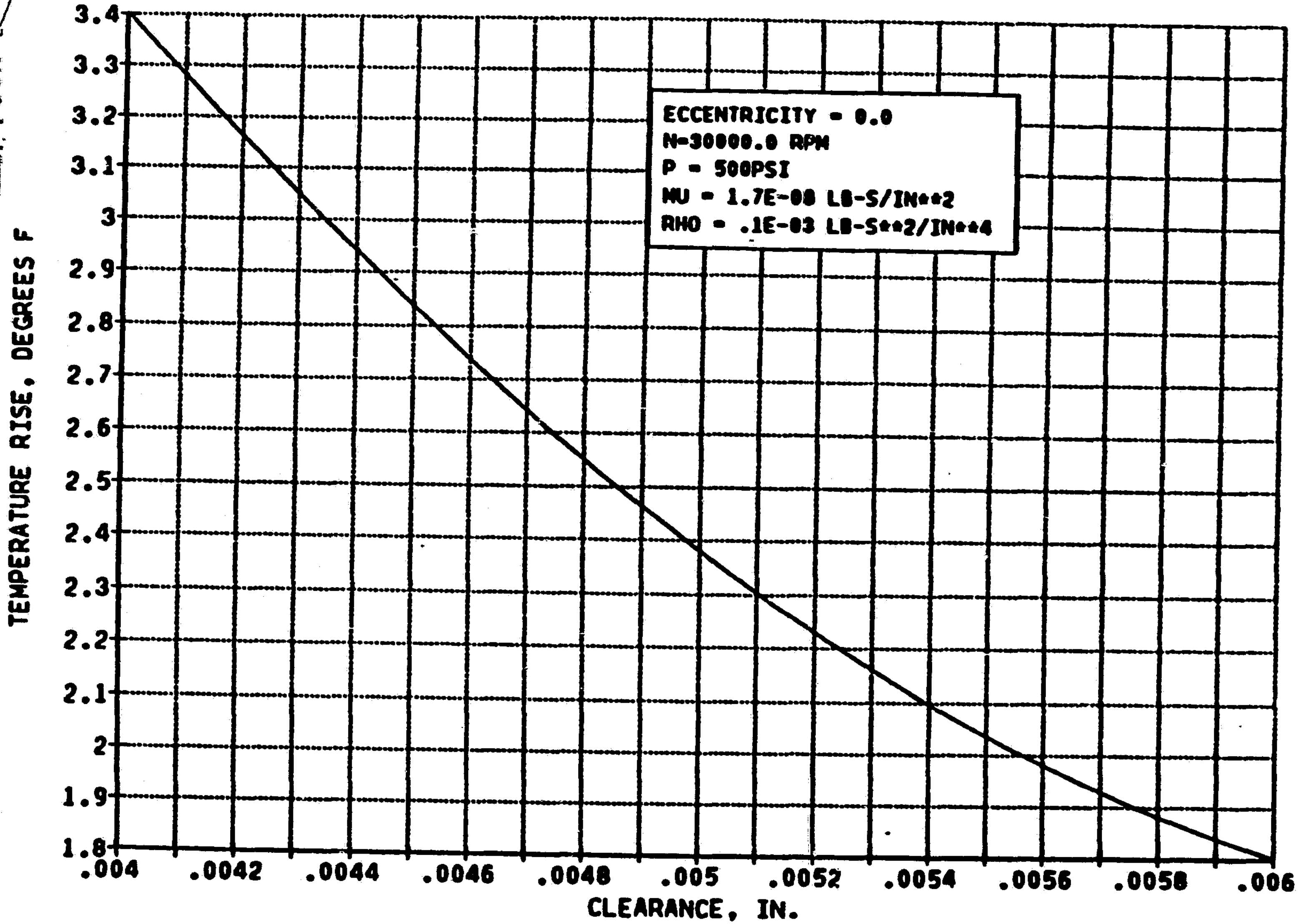

ΔW= -4.8%


VARIATION OF FLOW VS. CLEARANCE
4.01 lbs/s
2.97 lbs/s
ECCENTRICITY = 0.0
N=30000.0 RPM
P = 500PSI
MU = 1.7E-08 LB-S/IN**2
RHO = .1E-03 LB-S**2/IN**4
ΔQ = +87.4 %
FLOW, GPM
27
26
25
24
23
22
21
20
19
18
17
16
15
14
.004
.0042
.0044
.0046
.0048
.005
.0052
.0054
.0056
.0058
.006
CLEARANCE, IN.

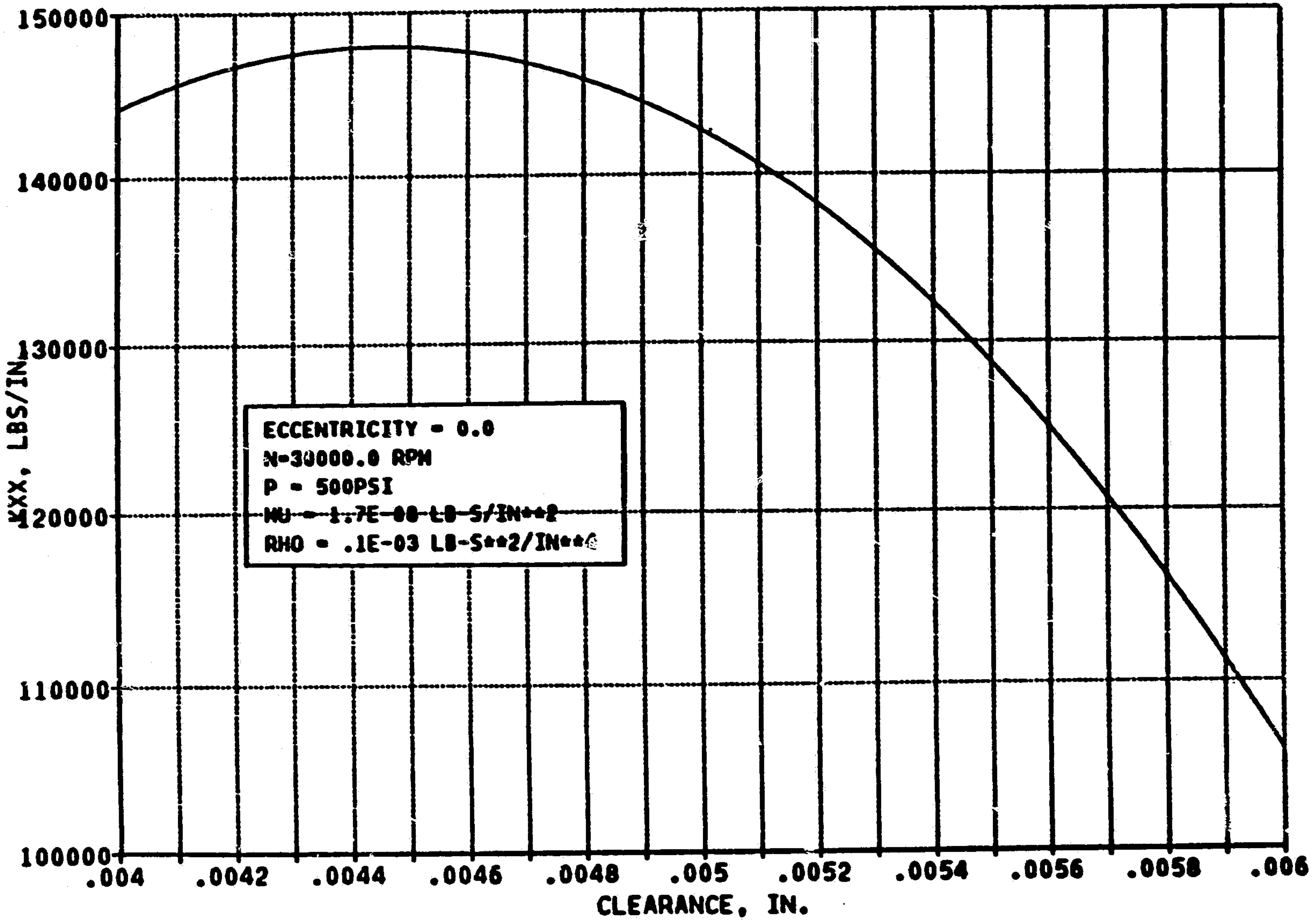


VARIATION OF POWER LOSS VS. CLEARANCE
ECCENTRICITY = 0.0
N=30000.0 RPM
P = 500PSI
MU = 1.7E-08 LB-S/IN**2
RHO = .1E-03 LB-S**2/IN**4
POWER LOSS, HP
6.04
6.02
6
5.98
5.96
5.94
5.92
5.9
5.88
5.86
5.84
5.82
5.8
5.78
5.76
5.74
5.72
5.7
5.68
5.66
5.64
5.62
5.6
5.58
.004
.0042
.0044
.0046
.0048
.005
.0052
.0054
.0056
.0058
.006
CLEARANCE, IN.

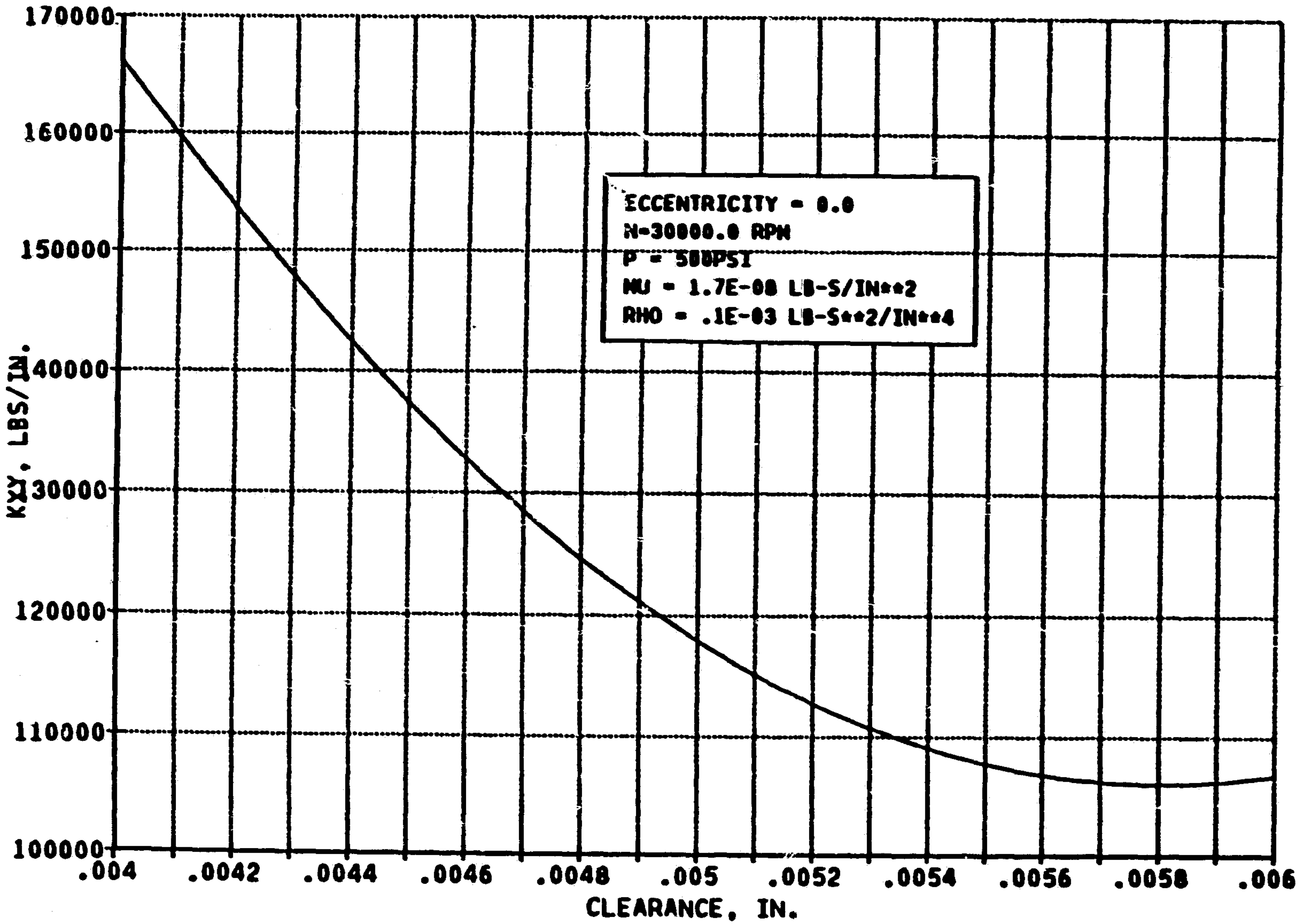
VARIATION OF TEMPERATURE RISE VS. CLEARANCE

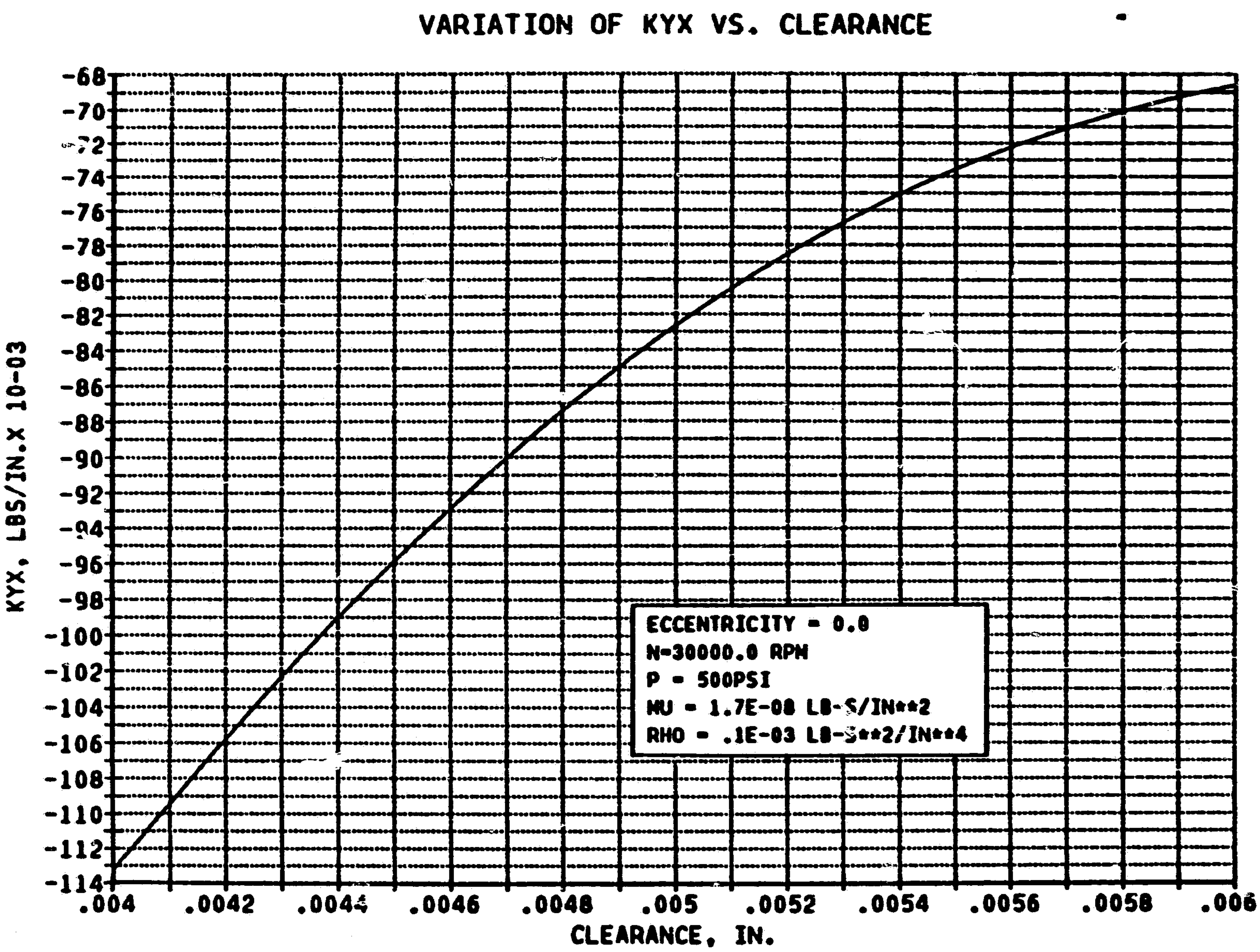


VARIATION OF KXX VS. CLEARANCE



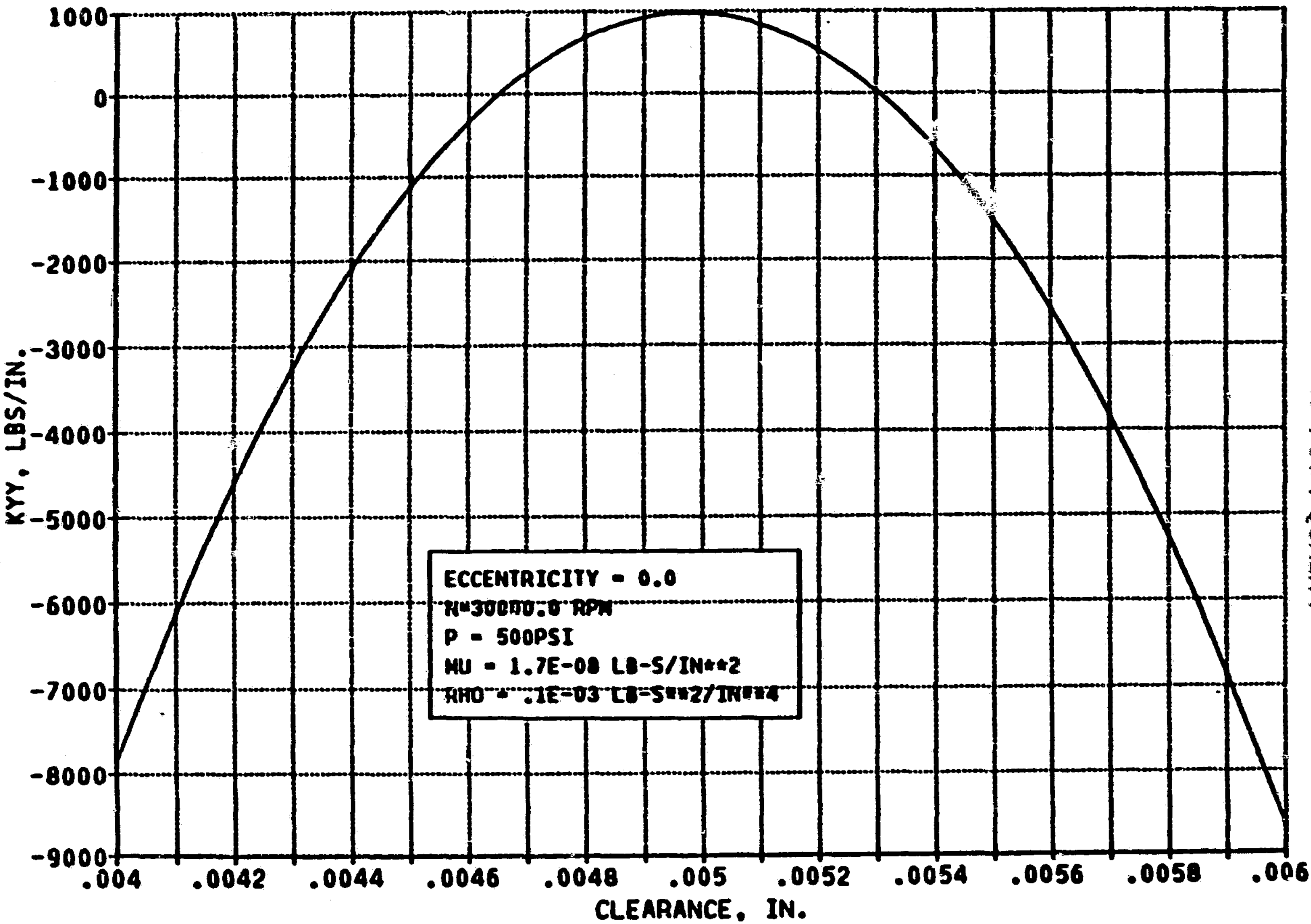
VARIATION OF KXY VS. CLEARANCE



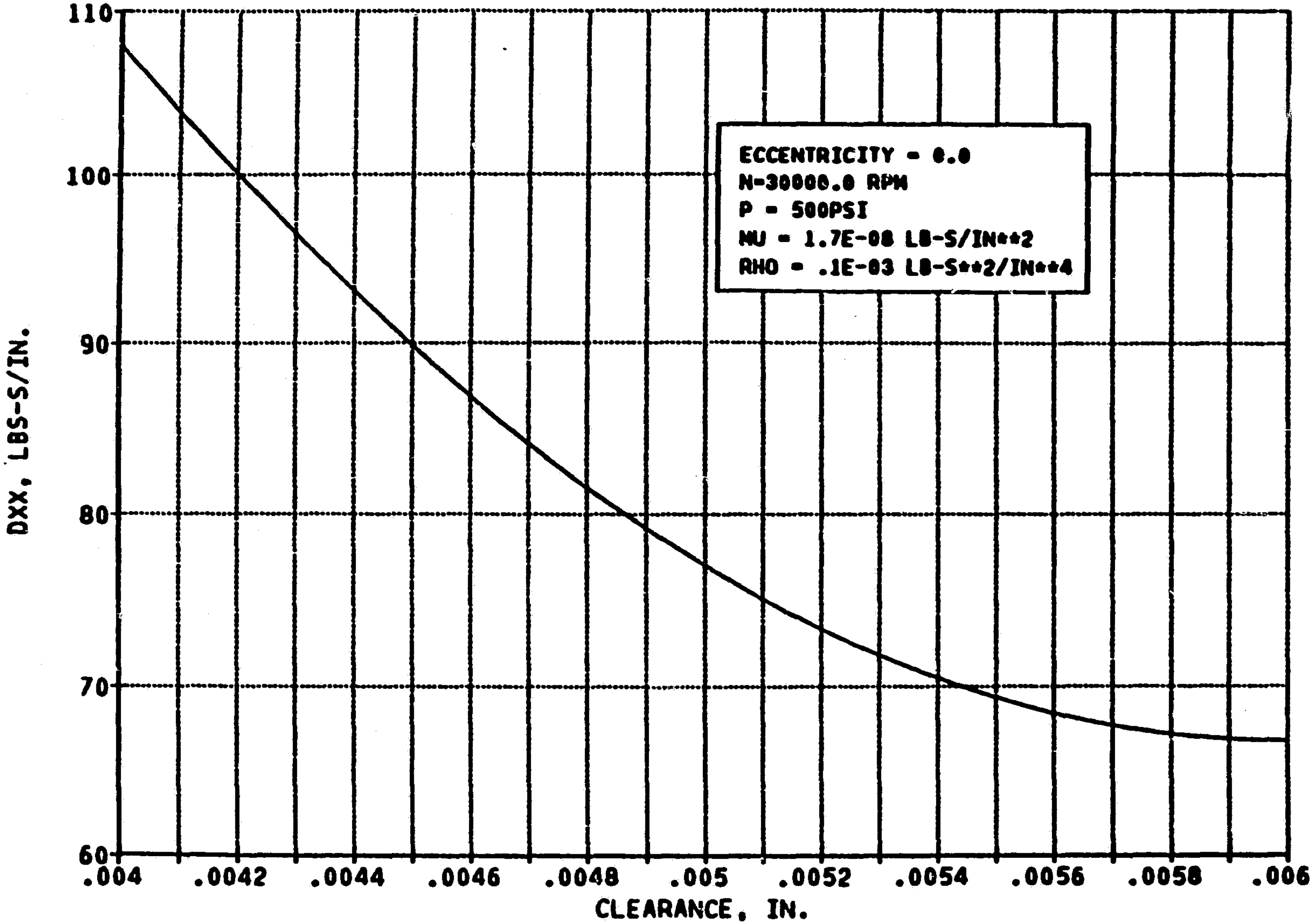


VARIATION OF KYX VS. CLEARANCE
KYX, LBS/IN.X 10-03
-68
-70
-72
-74
-76
-78
-80
-82
-84
-86
-88
-90
-92
-94
-96
-98
-100
-102
-104
-106
-108
-110
-112
-114
.004
.0042
.0044
.0046
.0048
.005
.0052
.0054
.0056
.0058
.006
CLEARANCE, IN.
ECCENTRICITY = 0.0
N=30000.0 RPM
P = 500PSI
MU = 1.7E-08 LB-S/IN**2
RHO = .1E-03 LB-S**2/IN**4

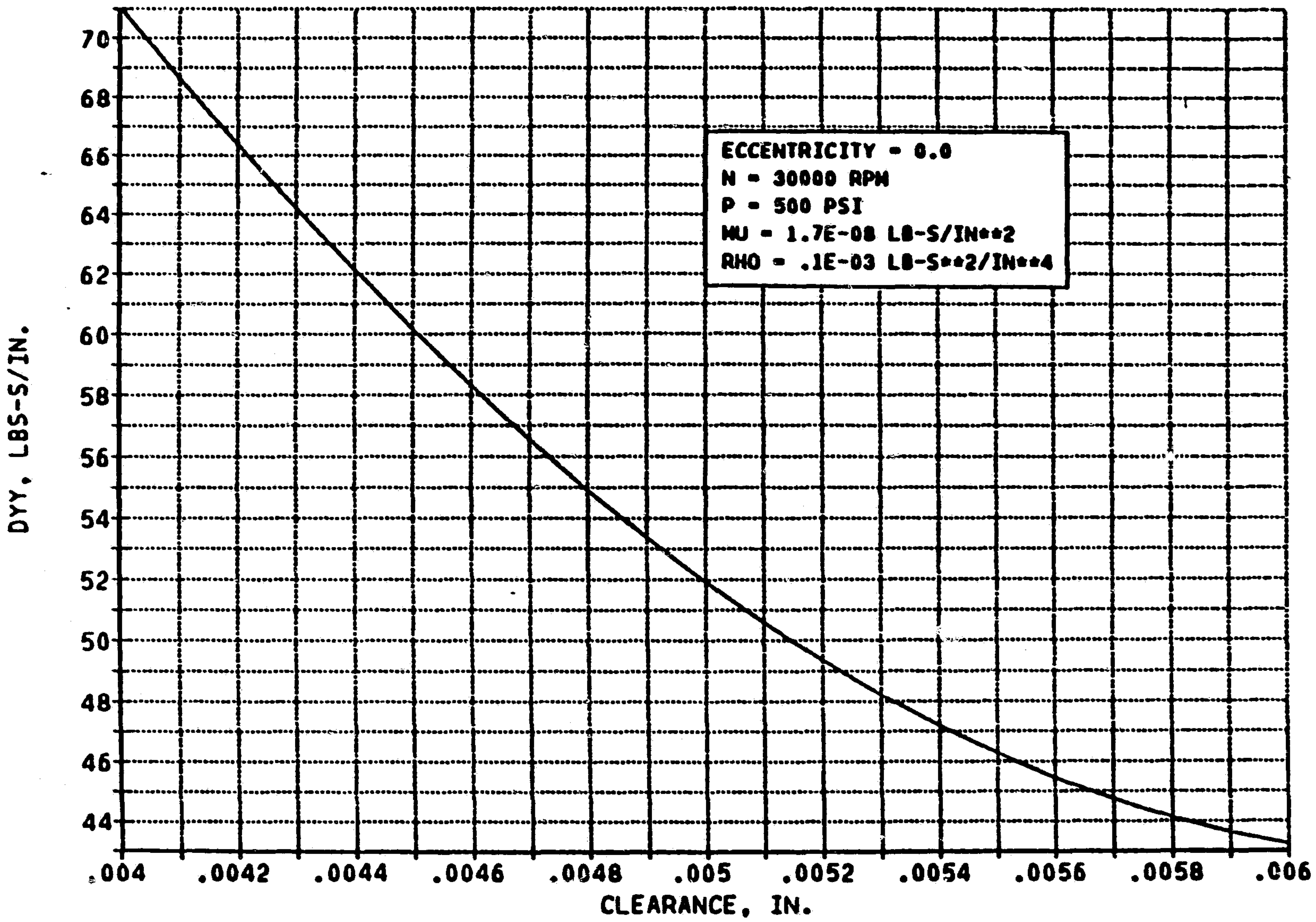
VARIATION OF KYY VS. CLEARANCE



VARIATION OF DXX VS. CLEARANCE



VARIATION OF DYY VS CLEARANCE



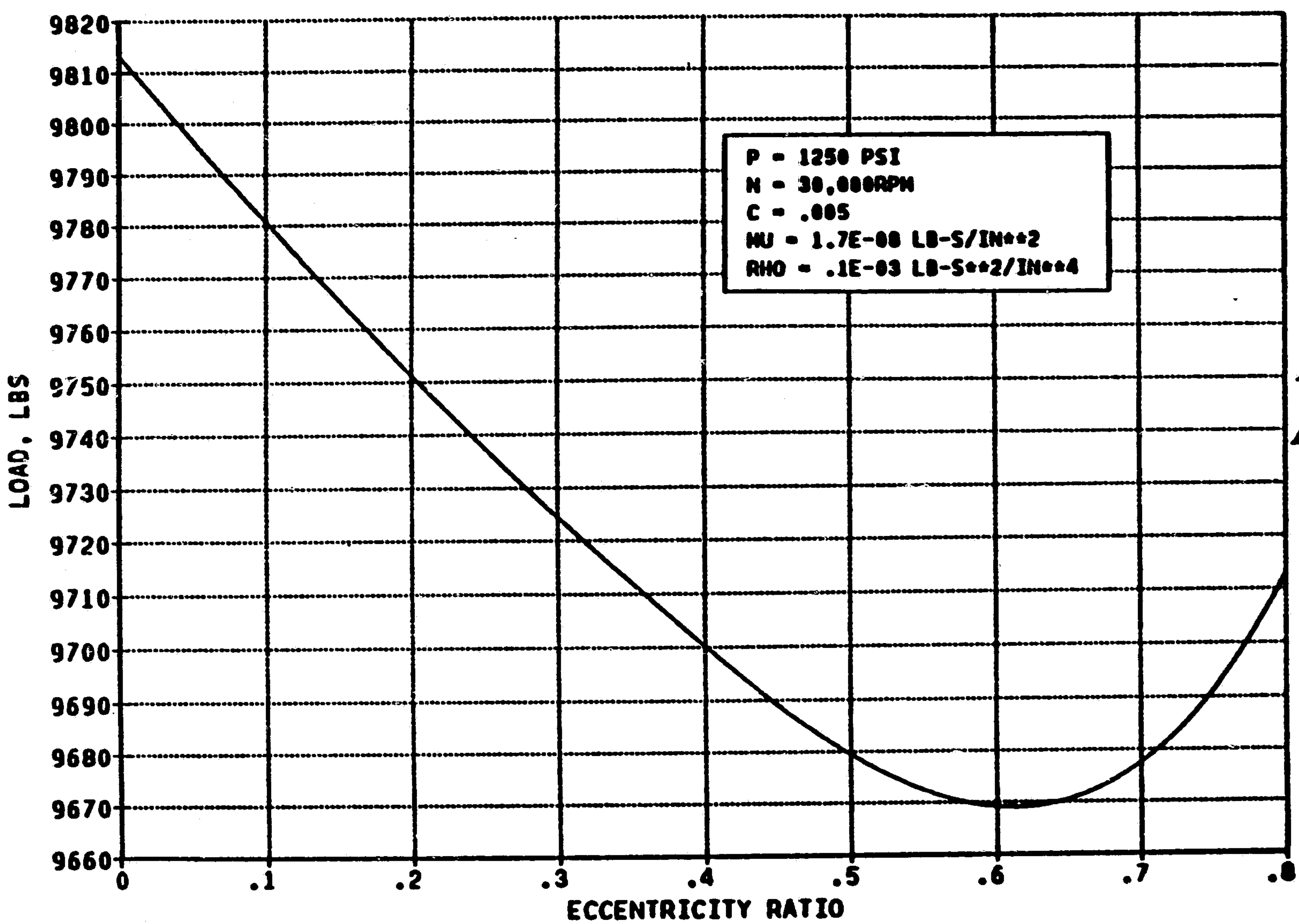
## 3.4 VARIATION OF PERFORMANCE WITH ECCENTRICITY RATIO

P = 1250 PSI

N = 30,000 RPM

C = 0.005

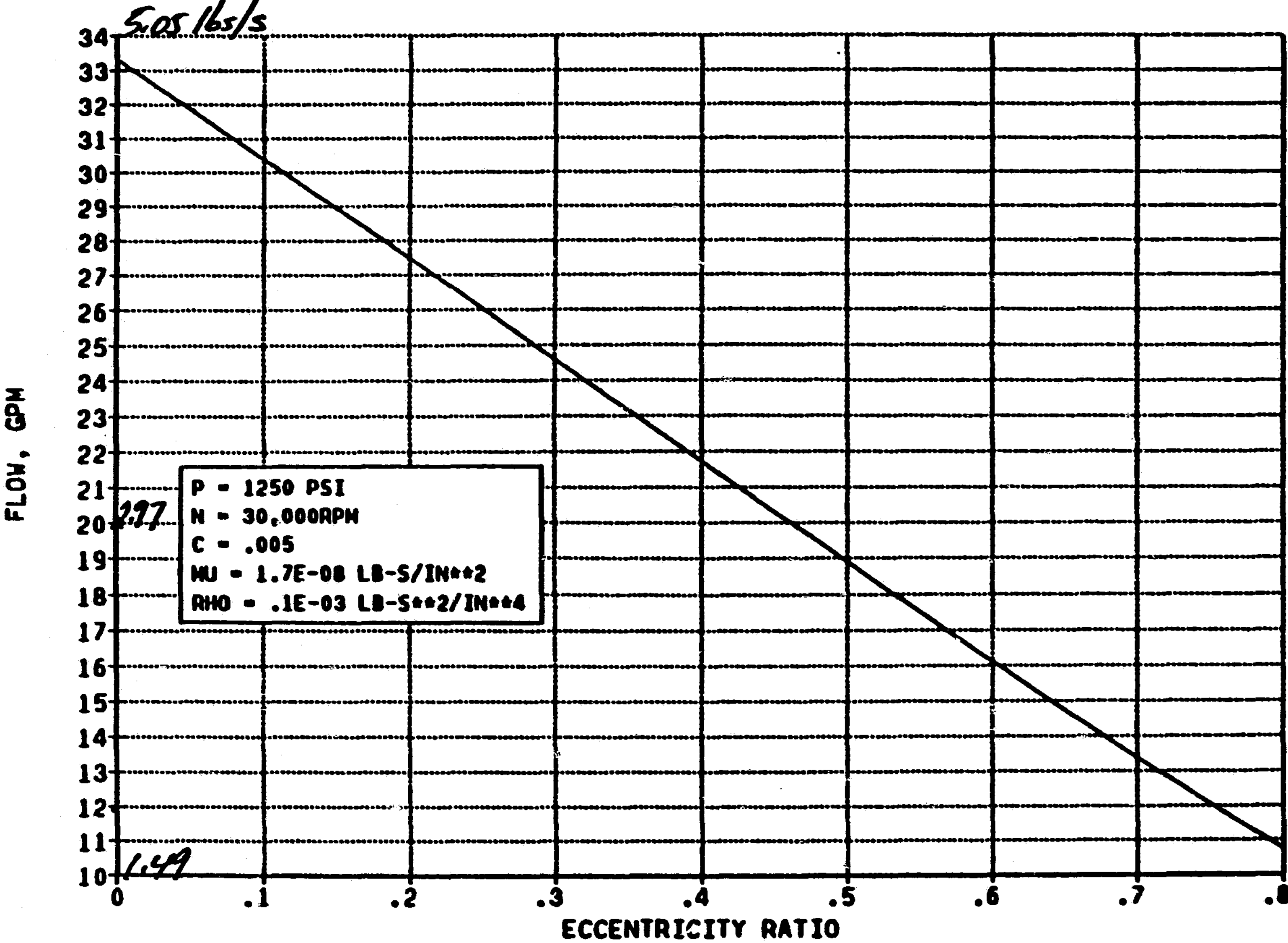
HYDROSTATIC LOADER, LOAD VS. ECCENTRICITY RATIO




$\Delta\varepsilon$ 0 → .4

$\Delta W = -1.14\%$

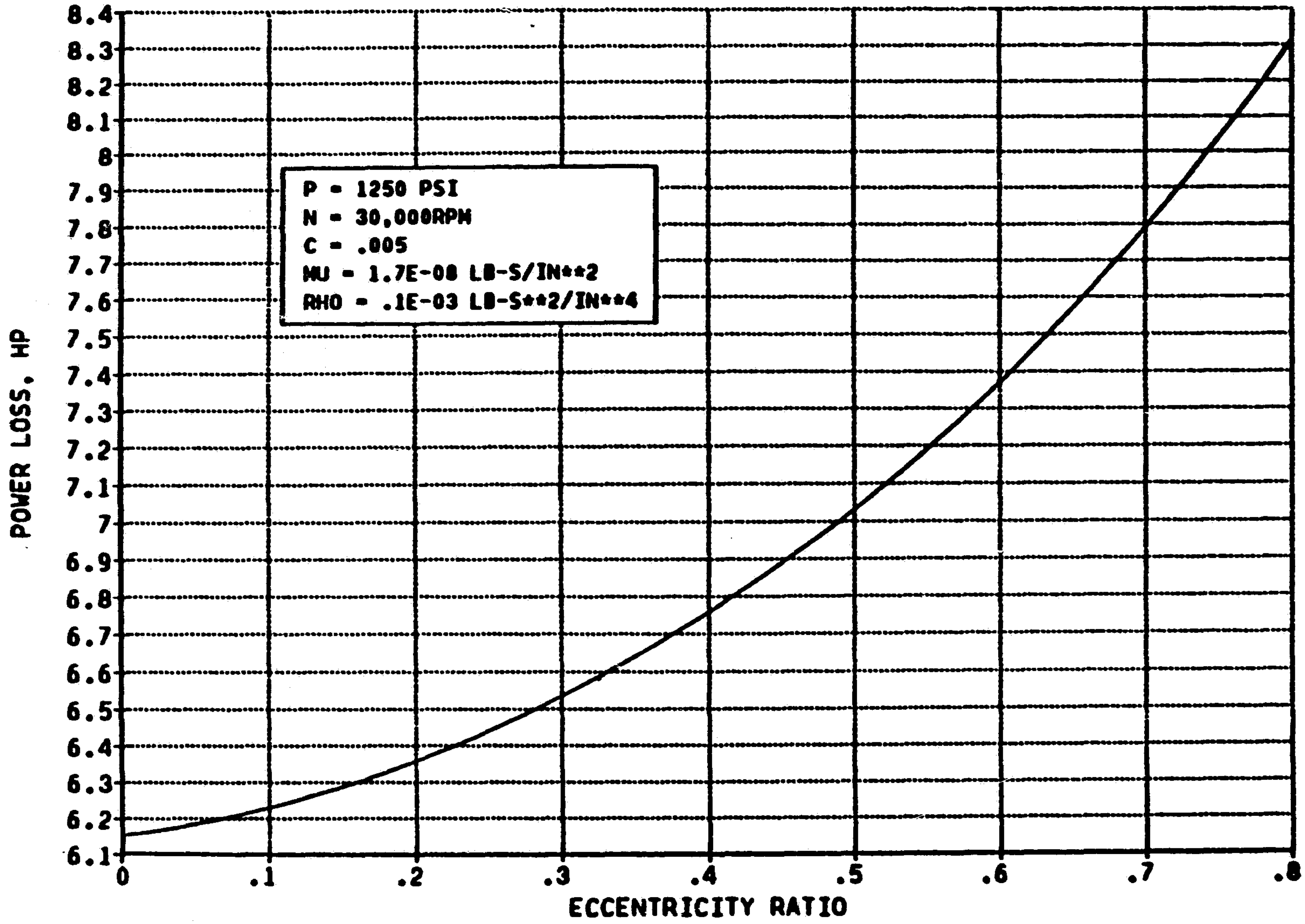
HYDROSTATIC LOADER, FLOW VS. ECCENTRICITY RATIO

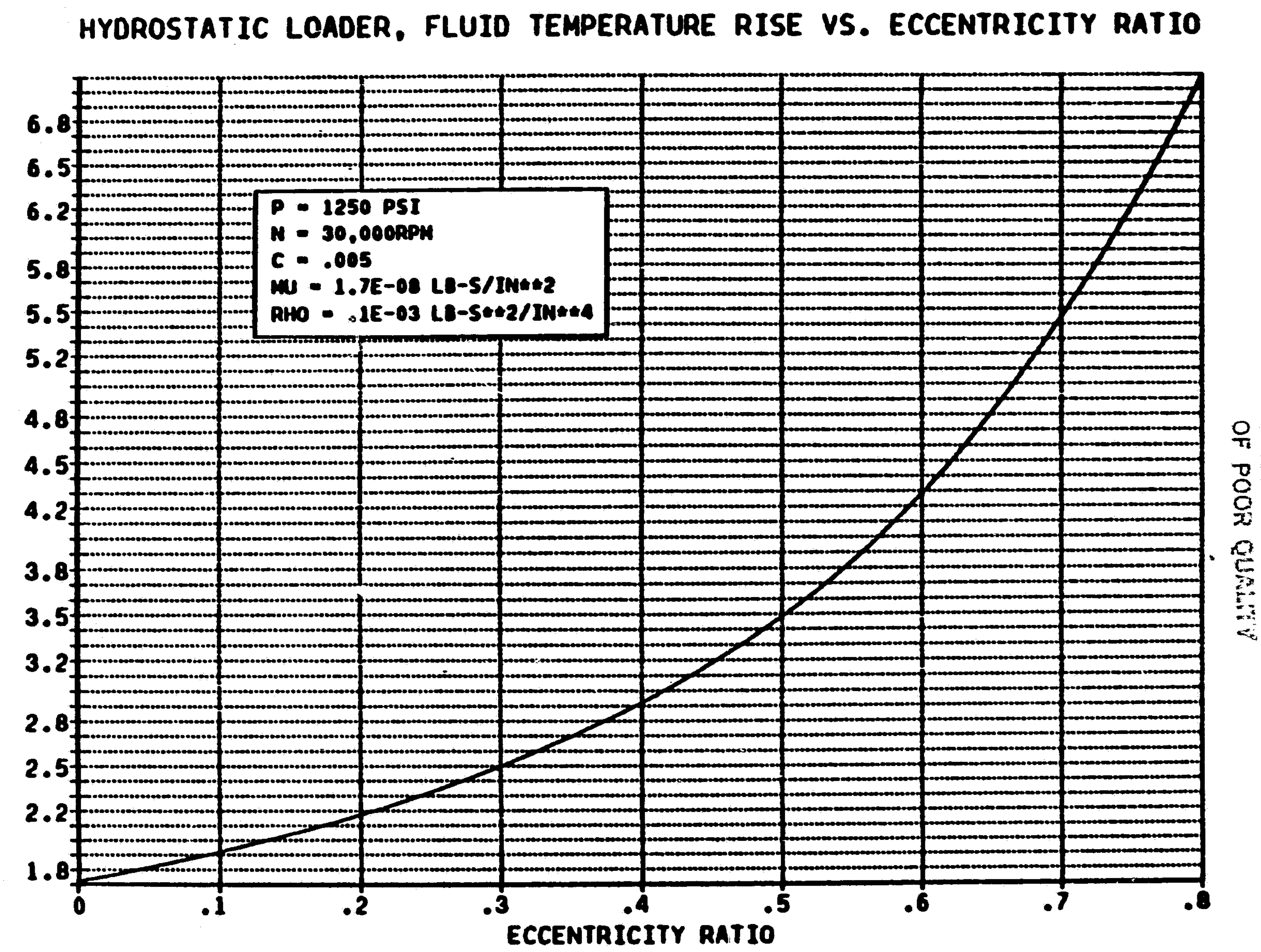



$\Delta E = 0 \rightarrow .4$

$\Delta Q = -34.3\%$

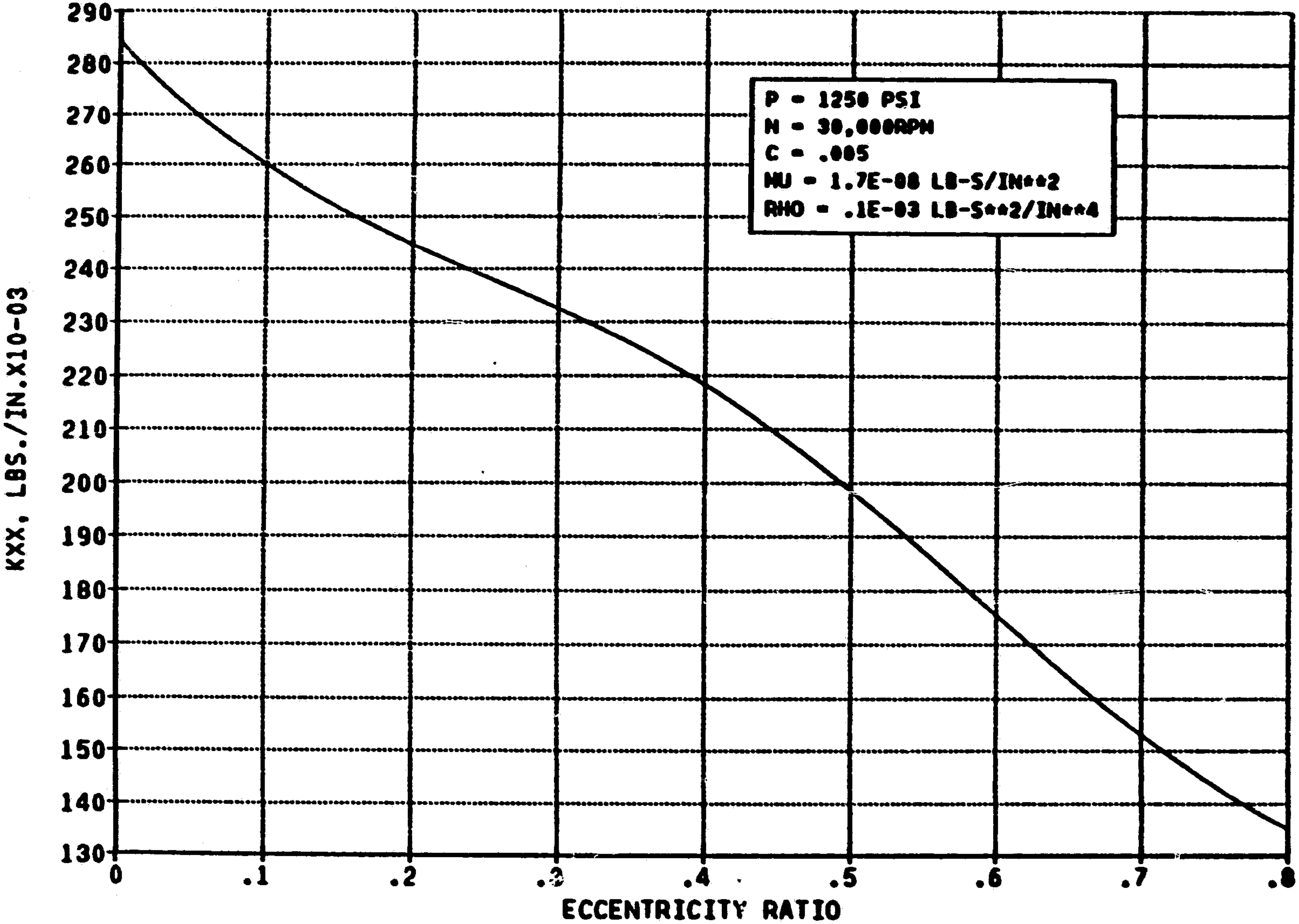
HYDROSTATIC LOADER, POWER LOSS VS. ECCENTRICITY RATIO



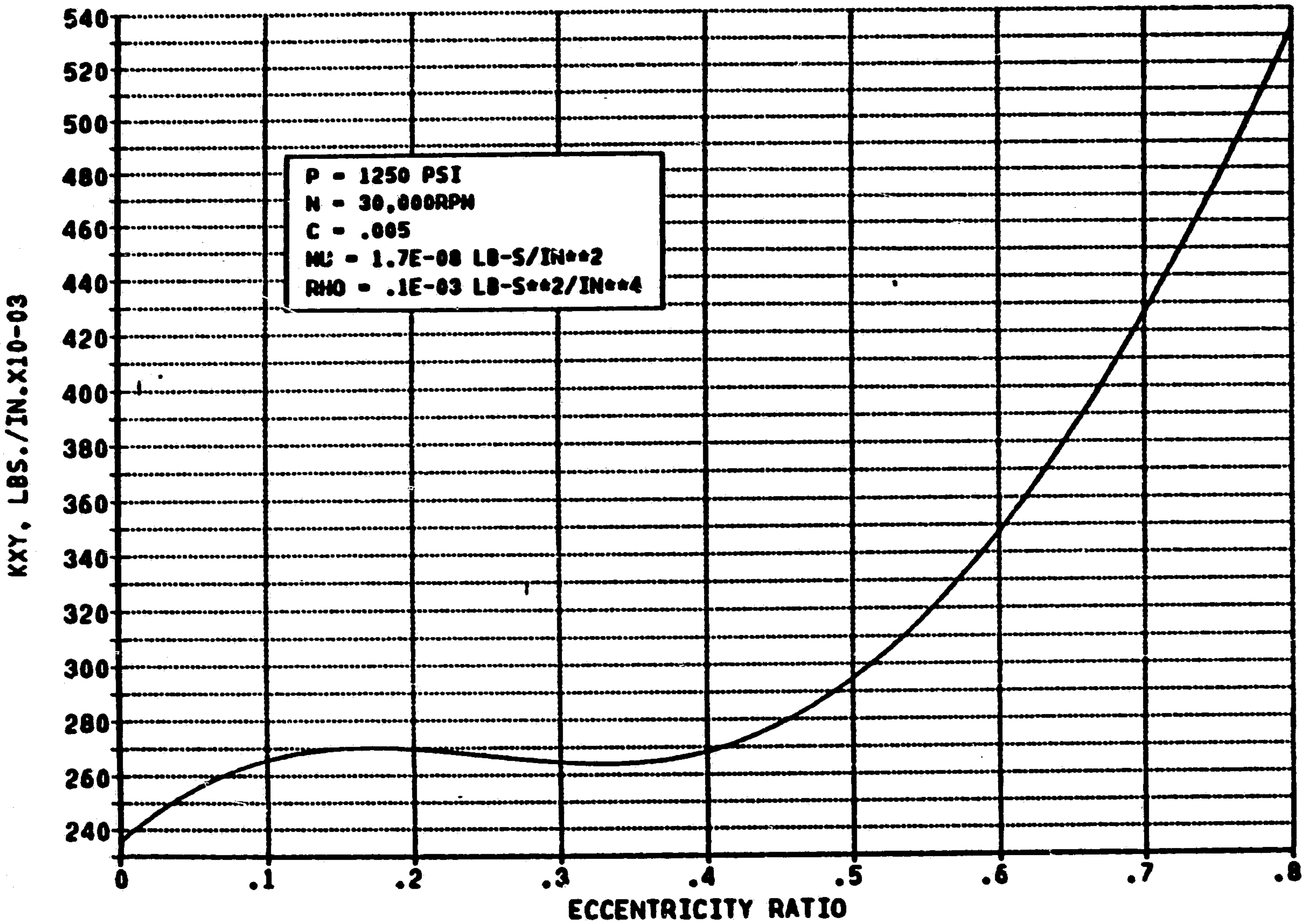


HYDROSTATIC LOADER, FLUID TEMPERATURE RISE VS. ECCENTRICITY RATIO
P = 1250 PSI
N = 30,000RPM
C = .005
MU = 1.7E-08 LB-S/IN**2
RHO = .1E-03 LB-S**2/IN**4
FLUID TEMPERATURE RISE, DEGREES F
6.8
6.5
6.2
5.8
5.5
5.2
4.8
4.5
4.2
3.8
3.5
3.2
2.8
2.5
2.2
1.8
0
.1
.2
.3
.4
.5
.6
.7
.8
ECCENTRICITY RATIO

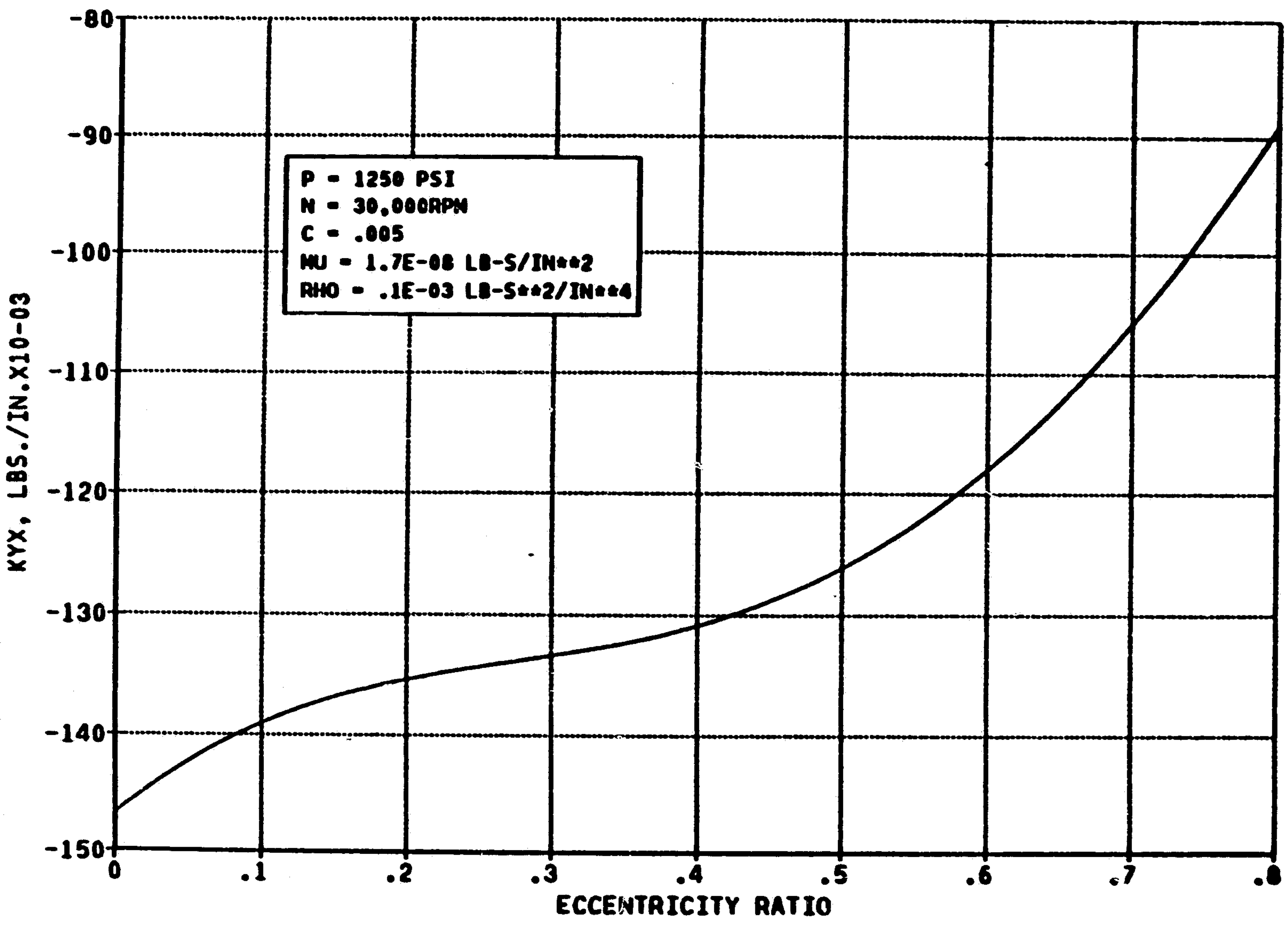
HYDROSTATIC LOADER, KXX VS. ECCENTRICITY RATIO



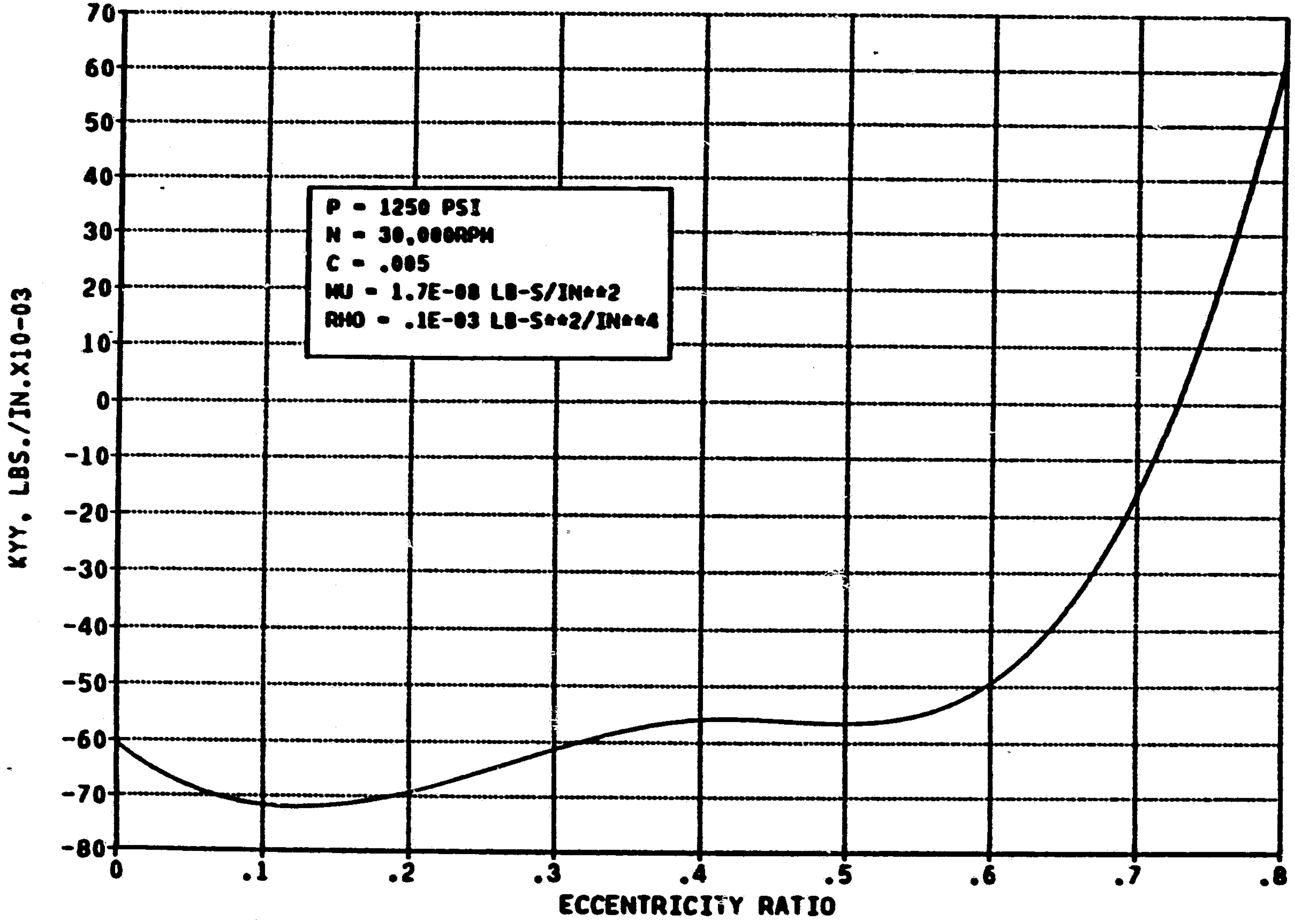
HYDROSTATIC LOADER, KXY VS. ECCENTRICITY RATIO



HYDROSTATIC LOADER, KYX VS. ECCENTRICITY RATIO

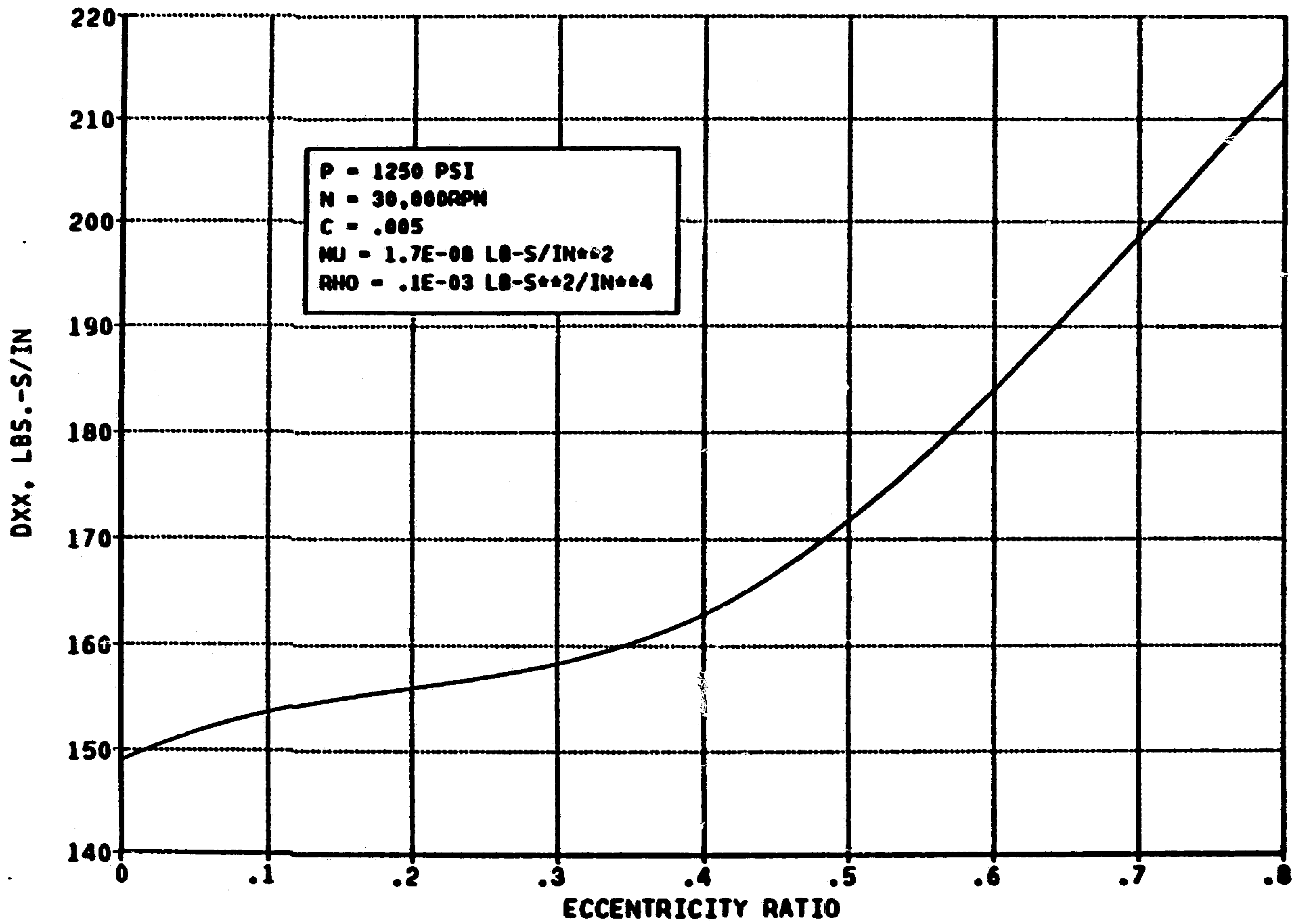


HYDROSTATIC LOADER, KYY VS. ECCENTRICITY RATIO

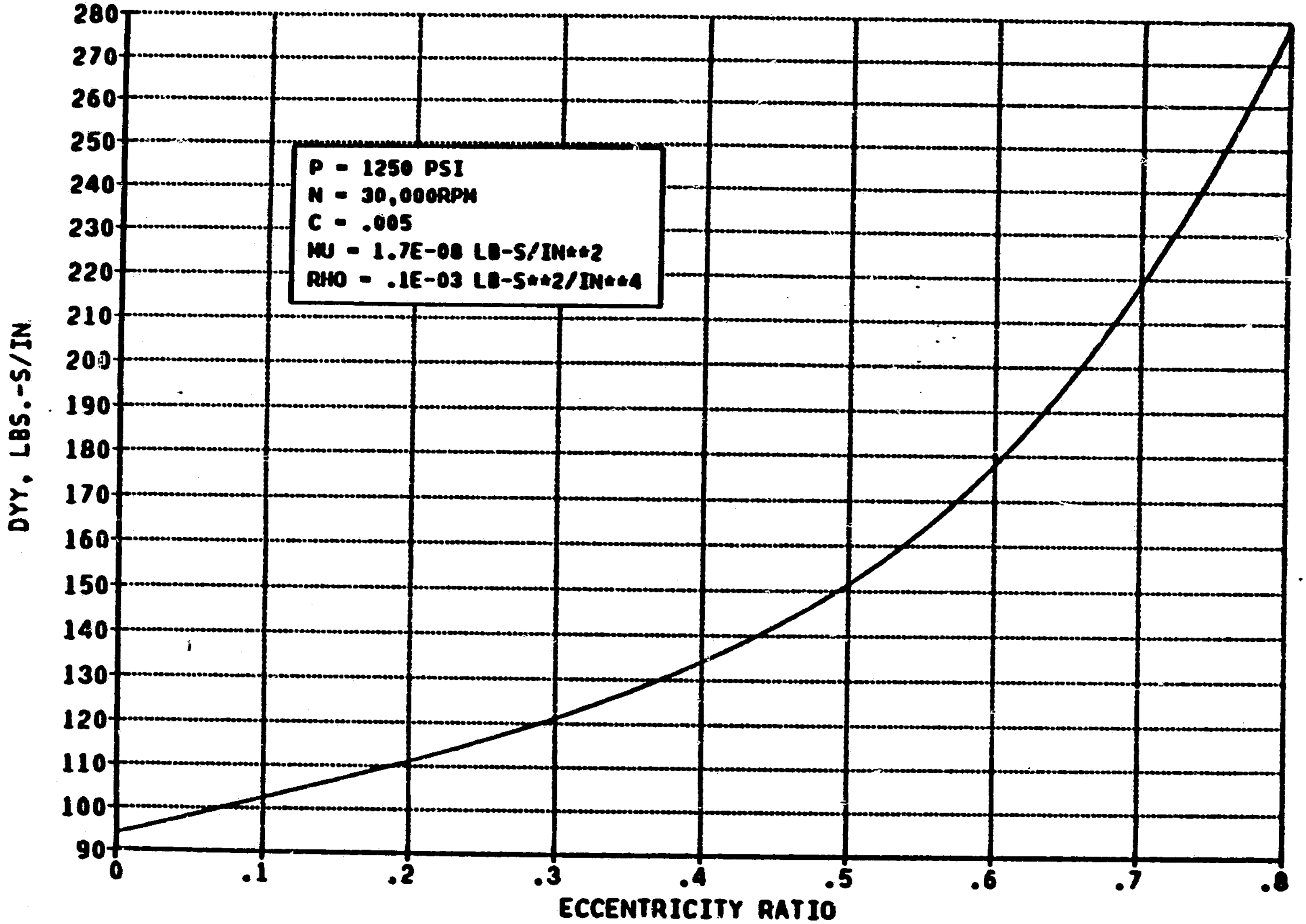


ORIGINAL PAGE IS OF POOR QUALITY

# HYDROSTATIC LOADER, DXX VS. ECCENTRICITY RATIO

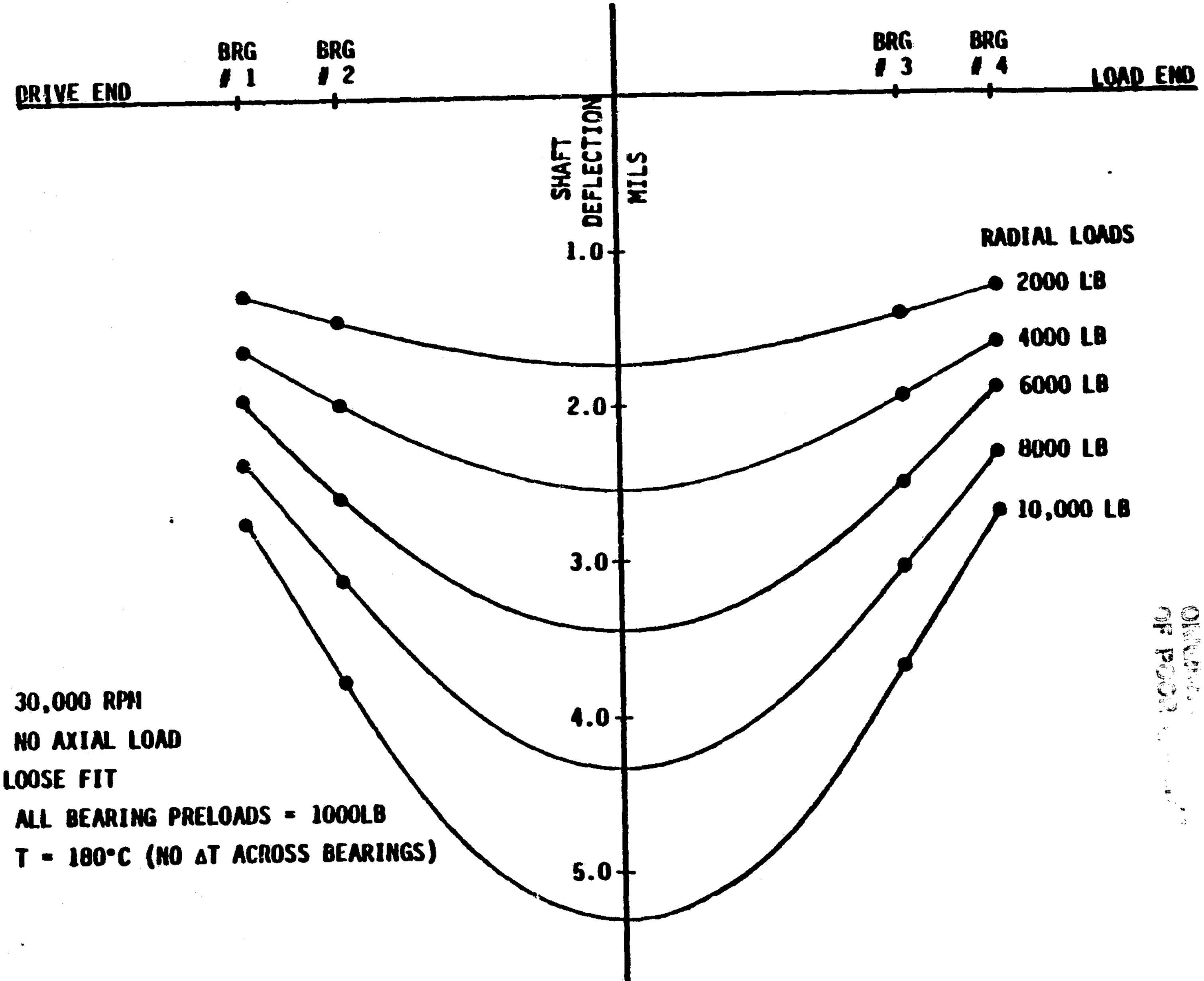


HYDROSTATIC LOADER, DYY VS. ECCENTRICITY RATIO



## 3.5 EFFECT OF SHAFT BENDING

# SHAFT DEFLECTIONS FOR LOX TESTER




30,000 RPM

NO AXIAL LOAD

LOOSE FIT

ALL BEARING PRELOADS = 1000LB

T = 180°C (NO ΔT ACROSS BEARINGS)

SRS

## EFFECT OF SHAFT BENDING

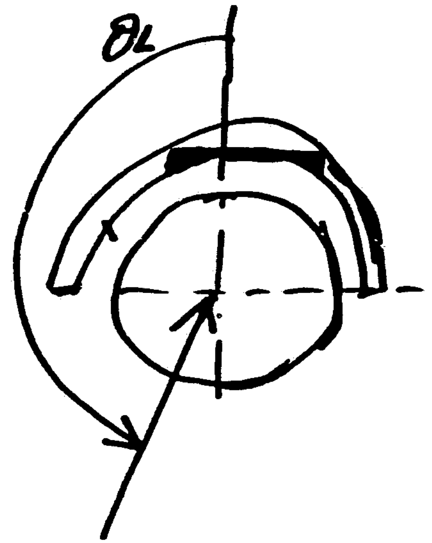
| p psi | $\epsilon$ --- | $\alpha$ Rad | W lbs | Q gpm | $P_L$ psi | $\Delta t$ °F | $K_{xx}$ | $K_{xy}$ | $K_{yx}$ | $K_{yy}$ | $D_{xx}$ lb-s/in | $D_{yy}$ lb-s/in | $\theta_L$ deg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | ← lb/in x $10^{-3}$ → | | | | | | |
| 1250 | 0 | 0 | 9813 | 33.3 | 6.13 | 1.72 | 284 | 236 | -147 | -61 | 149 | 94 | 180 |
| 1250 | 1.06 | .00096 | 10180 | 56.4 | 6.45 | 1.07 | 379 | 125 | -163 | -7.6 | 112 | 60 | 176 |
| 500 | 0 | 0 | 3873 | 19.8 | 5.77 | 2.38 | 143 | 118 | - 83 | .97 | 77 | 52 | 180 |
| 500 | 0.5 | .00032 | 4036 | 28.2 | 5.38 | 1.64 | 158 | 106 | - 89 | -7.7 | 72 | 43 | 176 |

Flow increase  p = 1250,  69.4%

p = 500,  42.4%

Load increase  p = 1250,  3.74%

p = 500,  4.2%




N = 30,000 rpm

C = 0.005 in.

$\theta_L$ = Load angle

## 3.6 WALOWIT APPROXIMATE ANALYSIS

SMOOTH,CR=.394MM

| | |
|---|---|
| 180.0 | ANGLE SUBTENDED BY LOADER (DEG) |
| 2.5 | DIAMETER (IN) |
| 5.0 | AXIAL LENGTH (IN) |
| 1.0355 | CIRCUMFERENTIAL LAND WIDTH |
| 0.88 | AXIAL LAND WIDTH (IN) |
| 1250.0 | RECESS PRESSURE (PSI) |
| 0.5000000E-02 | CLEARANCE (IN) |
| 0.1000000E-03 | DENSITY (LB-SEC**2/IN**4) |
| 0.1700000E-07 | VISCOSITY (LB-SEC/IN**2) |
| -0.21663 | MS (CHILDS EXPONENT) |
| 0.06736 | NS (CHILDS COEFFICIENT) |
| 0.0311894 | G FACTOR OVER CIRCUMFERENTIAL LAND |
| 0.0299025 | G FACTOR OVER AXIAL LAND |
| 175155.0 | MINIMUM REYNOLDS NUMBER |
| 806.686 | PRESSURE RISE OVER CIRCUMFERENTIAL LAND (PSI) |
| 755.331 | PRESSURE RISE OVER AXIAL LAND (PSI) |
| 9729.79 | LOAD (LB) |
| 154.853 | FLOW (IN**3/SEC) |
| 203946.0 | VERTICAL STIFFNESS (LB/IN) |
| 251432.0 | HORIZONTAL STIFFNESS (LB/IN) |

SMOOTH, CR=.527MM

| | |
|---|---|
| 180.0 | ANGLE SUBTENDED BY LOADER (DEG) |
| 2.5 | DIAMETER (IN) |
| 5.0 | AXIAL LENGTH (IN) |
| 1.0355 | CIRCUMFERENTIAL LAND WIDTH |
| 0.88 | AXIAL LAND WIDTH (IN) |
| 1250.0 | RECESS PRESSURE (PSI) |
| 0.5000000E-02 | CLEARANCE (IN) |
| 0.1000000E-03 | DENSITY (LB-SEC**2/IN**4) |
| 0.1700000E-07 | VISCOSITY (LB-SEC/IN**2) |
| -0.2398 | MS (CHILDS EXPONENT) |
| 0.09885 | NS (CHILDS COEFFICIENT) |
| 0.0298221 | G FACTOR OVER CIRCUMFERENTIAL LAND |
| 0.0285748 | G FACTOR OVER AXIAL LAND |
| 171390.0 | MINIMUM REYNOLDS NUMBER |
| 825.538 | PRESSURE RISE OVER CIRCUMFERENTIAL LAND (PSI) |
| 774.754 | PRESSURE RISE OVER AXIAL LAND (PSI) |
| 9800.26 | LOAD (LB) |
| 151.622 | FLOW (IN**3/SEC) |
| 203856.0 | VERTICAL STIFFNESS (LB/IN) |
| 250778.0 | HORIZONTAL STIFFNESS (LB/IN) |

ROCKETDYNE

| | |
|---|---|
| 180.0 | ANGLE SUBTENDED BY LOADER (DEG) |
| 2.5 | DIAMETER (IN) |
| 5.0 | AXIAL LENGTH (IN) |
| 1.0355 | CIRCUMFERENTIAL LAND WIDTH |
| 0.88 | AXIAL LAND WIDTH (IN) |
| 1250.0 | RECESS PRESSURE (PSI) |
| 0.5000000E-02 | CLEARANCE (IN) |
| 0.1000000E-03 | DENSITY (LB-SEC**2/IN**4) |
| 0.1700000E-07 | VISCOSITY (LB-SEC/IN**2) |
| -0.13567 | MS (CHILDS EXPONENT) |
| 0.06968 | NS (CHILDS COEFFICIENT) |
| 0.0196034 | G FACTOR OVER CIRCUMFERENTIAL LAND |
| 0.0185269 | G FACTOR OVER AXIAL LAND |
| 134771.0 | MINIMUM REYNOLDS NUMBER |
| 987.541 | PRESSURE RISE OVER CIRCUMFERENTIAL LAND (PSI) |
| 949.765 | PRESSURE RISE OVER AXIAL LAND (PSI) |
| 10419.7 | LOAD (LB) |
| 119.712 | FLOW (IN**3/SEC) |
| 142017.0 | VERTICAL STIFFNESS (LB/IN) |
| 171361.0 | HORIZONTAL STIFFNESS (LB/IN) |

DIAMOND GRID

| | |
|---|---|
| 180.0 | ANGLE SUBTENDED BY LOADER (DEG) |
| 2.5 | DIAMETER (IN) |
| 5.0 | AXIAL LENGTH (IN) |
| 1.0355 | CIRCUMFERENTIAL LAND WIDTH |
| 0.88 | AXIAL LAND WIDTH (IN) |
| 1250.0 | RECESS PRESSURE (PSI) |
| 0.5000000E-02 | CLEARANCE (IN) |
| 0.1000000E-03 | DENSITY (LB-SEC**2/IN**4) |
| 0.1700000E-07 | VISCOSITY (LB-SEC/IN**2) |
| -0.03498 | MS (CHILDS EXPONENT) |
| 0.11815 | NS (CHILDS COEFFICIENT) |
| 0.8304740E-02 | G FACTOR OVER CIRCUMFERENTIAL LAND |
| 0.7706740E-02 | G FACTOR OVER AXIAL LAND |
| 68363.7 | MINIMUM REYNOLDS NUMBER |
| 1182.47 | PRESSURE RISE OVER CIRCUMFERENTIAL LAND (PSI) |
| 1171.02 | PRESSURE RISE OVER AXIAL LAND (PSI) |
| 11183.5 | LOAD (LB) |
| 60.9807 | FLOW (IN**3/SEC) |
| 38973.7 | VERTICAL STIFFNESS (LB/IN) |
| 46021.6 | HORIZONTAL STIFFNESS (LB/IN) |

HOLE PATTERN

| | |
|---|---|
| 180.0 | ANGLE SUBTENDED BY LOADER (DEG) |
| 2.5 | DIAMETER (IN) |
| 5.0 | AXIAL LENGTH (IN) |
| 1.0355 | CIRCUMFERENTIAL LAND WIDTH |
| 0.88 | AXIAL LAND WIDTH (IN) |
| 1250.0 | RECESS PRESSURE (PSI) |
| 0.5000000E-02 | CLEARANCE (IN) |
| 0.1000000E-03 | DENSITY (LB-SEC**2/IN**4) |
| 0.1700000E-07 | VISCOSITY (LB-SEC/IN**2) |
| 0.01904 | MS (CHILDS EXPONENT) |
| 0.015027 | NS (CHILDS COEFFICIENT) |
| 0.0173207 | G FACTOR OVER CIRCUMFERENTIAL LAND |
| 0.0162356 | G FACTOR OVER AXIAL LAND |
| 123954.0 | MINIMUM REYNOLDS NUMBER |
| 1027.98 | PRESSURE RISE OVER CIRCUMFERENTIAL LAND (PSI) |
| 996.292 | PRESSURE RISE OVER AXIAL LAND (PSI) |
| 10579.3 | LOAD (LB) |
| 110.082 | FLOW (IN**3/SEC) |
| 105897.0 | VERTICAL STIFFNESS (LB/IN) |
| 127372.0 | HORIZONTAL STIFFNESS (LB/IN) |

## RESULTS OF WALOWIT APPROXIMATE ANALYSIS

| Configuration | W lbs | Q gpm | $K_{yy}$ lbs/in x $10^{-3}$ | $K_{xx}$ lbs/in x $10^{-3}$ |
|---|---|---|---|---|
| Smooth CR = .394 mm | 9730 | 40.26 | 203.9 | 251.4 |
| Smooth CR = .527 mm | 9800 | 39.33 | 203.9 | 251 |
| Rocketdyne | 10420 | 31.13 | 142 | 171 |
| Diamond | 11183 | 15.85 | 39 | 46 |
| Hole | 10579 | 28.62 | 105.9 | 127.3 |

# 4.0 ROTOR DYNAMICS

```
***ROTOR DYNAMICS PROGRAM***

SEAL TESTER STABILITY WITHOUT LOADER, 4 BEARINGS

*FOUR DEGREE OF FREEDOM ANALYSIS*

NUMBER OF STATIONS.........  13
NUMBER OF DISCS............   8
NUMBER OF BEARINGS.........   2
NUMBER OF MATERIALS........   1
NUMBER OF SHAFT ELMTS......  12
GRAVITY LOADING............  NO
GRAVITY....................   386.4
SPEED...................... 30000.0
```

***ROTOR DYNAMICS PROGRAM***

SEAL TESTER STABILITY WITHOUT LOADER, 4 BEARINGS

* MATERIAL PROPERTIES *

| MATERIAL TYPE NO. | MODULUS OF ELASTICITY | SHEAR MODULUS | DENSITY | MATERIAL DAMPING |
|---|---|---|---|---|
| 1 | 0.30000000E+08 | 0.12000000E+08 | 0.76604565E-03 | 0.0 |

OF Po

***ROTOR DYNAMICS PROGRAM***

SEAL TESTER STABILITY WITHOUT LOADER, 4 BEARINGS

* SHAFT ELEMENT DATA *

| SHAFT ELEMENT | ELEMENT STATIONS | | ELEMENT TYPE | MATERIAL TYPE | X(1) | X(2) | X(3) | X(4) | X(5) |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1 | 2 | 1 | 1 | 0.968000E+00 | 0.226000E+01 | 0.0 | 0.226000E+01 | 0.0 |
| 2 | 2 | 3 | 2 | 1 | 0.225800E+01 | 0.226000E+01 | 0.150000E+01 | 0.226000E+01 | 0.150000E+01 |
| 3 | 3 | 4 | 2 | 1 | 0.149700E+01 | 0.226000E+01 | 0.150000E+01 | 0.226000E+01 | 0.150000E+01 |
| 4 | 4 | 5 | 2 | 1 | 0.161300E+01 | 0.226000E+01 | 0.150000E+01 | 0.226000E+01 | 0.150000E+01 |
| 5 | 5 | 6 | 2 | 1 | 0.258000E+01 | 0.250000E+01 | 0.150000E+01 | 0.250000E+01 | 0.150000E+01 |
| 6 | 6 | 7 | 2 | 1 | 0.258000E+01 | 0.250000E+01 | 0.150000E+01 | 0.250000E+01 | 0.150000E+01 |
| 7 | 7 | 8 | 2 | 1 | 0.161300E+01 | 0.226000E+01 | 0.150000E+01 | 0.226000E+01 | 0.150000E+01 |
| 8 | 8 | 9 | 2 | 1 | 0.148400E+01 | 0.226000E+01 | 0.150000E+01 | 0.226000E+01 | 0.150000E+01 |
| 9 | 9 | 10 | 2 | 1 | 0.187100E+01 | 0.226000E+01 | 0.150000E+01 | 0.226000E+01 | 0.150000E+01 |
| 10 | 10 | 11 | 2 | 1 | 0.387000E+00 | 0.226000E+01 | 0.150000E+01 | 0.226000E+01 | 0.150000E+01 |
| 11 | 11 | 12 | 2 | 1 | 0.452000E+00 | 0.226000E+01 | 0.150000E+01 | 0.226000E+01 | 0.150000E+01 |
| 12 | 12 | 13 | 2 | 1 | 0.774000E+00 | 0.122600E+01 | 0.645000E+00 | 0.122600E+01 | 0.645000E+00 |

X (1) = element length

X (2) = left side OD

X (3) = left side ID

X (4) = right side OD

X (5) = right side ID

***ROTOR DYNAMICS PROGRAM***

SEAL TESTER STABILITY WITHOUT LOADER, 4 BEARINGS

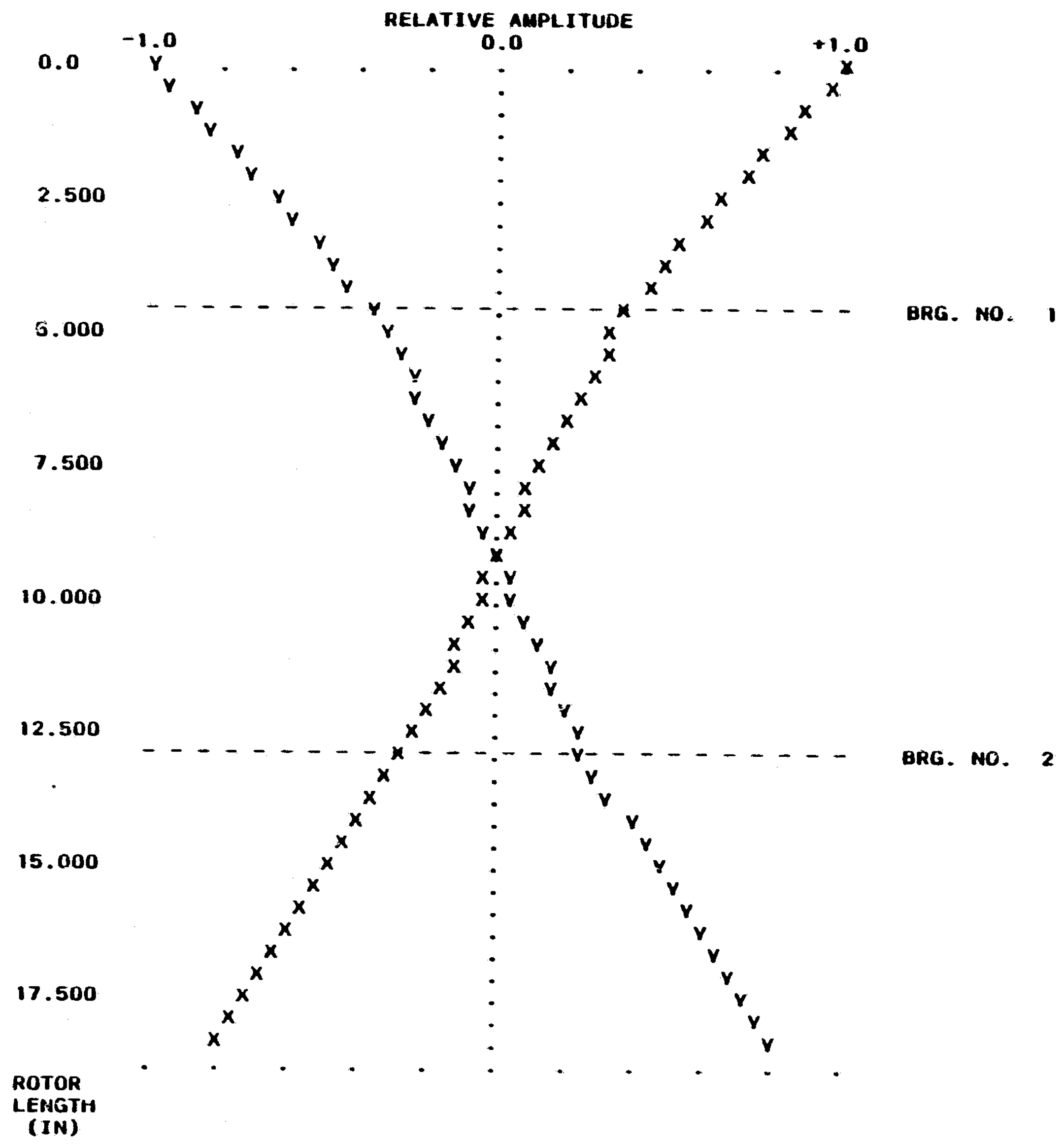
* ROTOR PARAMETERS *

| | |
|---|---|
| WEIGHT OF SHAFT............ | 13.56720 |
| WEIGHT OF DISCS............ | 8.85339 |
| WEIGHT OF ROTOR............ | 22.42058 |
| SHAFT LENGTH............... | 18.07697 |
| LOCATION OF C.G............ | 8.69985 |

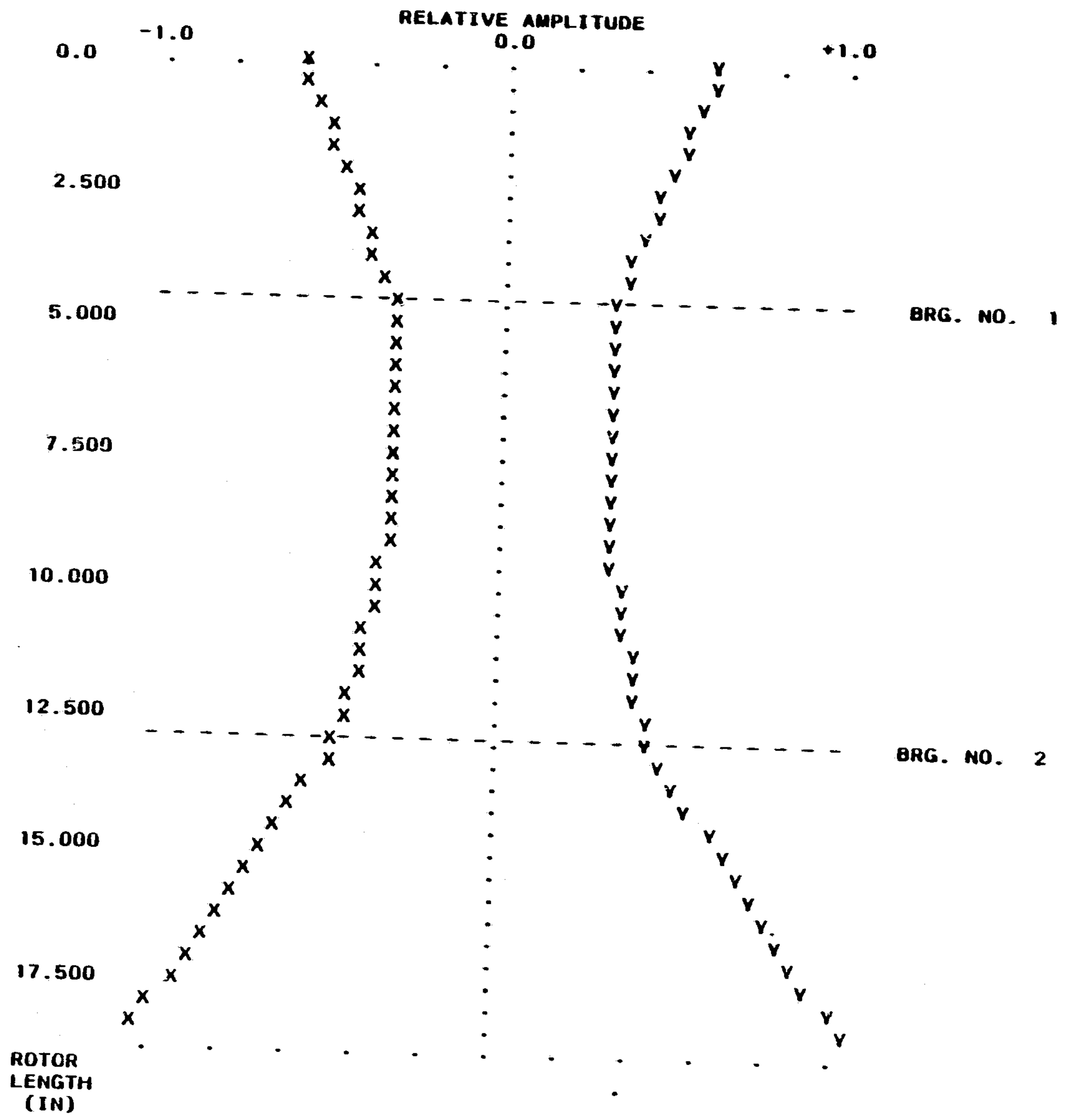
| DISC NUMBER | STATION NUMBER | POLAR MOMENT OF INERTIA | TRANSVERSE MOMENT OF INERTIA | MASS | RADIUS OF C.G. | ANGULAR POSITION OF C.G. |
|---|---|---|---|---|---|---|
| 1 | 2 | 0.939959E-02 | 0.469979E-02 | 0.312888E-02 | 0.0 | 0.0 |
| 2 | 3 | 0.175000E-02 | 0.875000E-03 | 0.282868E-02 | 0.0 | 0.0 |
| 3 | 4 | 0.310000E-02 | 0.155000E-02 | 0.407919E-02 | 0.0 | 0.0 |
| 4 | 5 | 0.129995E-02 | 0.650104E-03 | 0.120600E-02 | 0.0 | 0.0 |
| 5 | 7 | 0.129995E-02 | 0.650104E-03 | 0.120600E-02 | 0.0 | 0.0 |
| 6 | 8 | 0.310000E-02 | 0.155000E-02 | 0.407919E-02 | 0.0 | 0.0 |
| 7 | 9 | 0.175000E-02 | 0.875000E-03 | 0.282868E-02 | 0.0 | 0.0 |
| 8 | 10 | 0.364001E-02 | 0.181988E-02 | 0.355590E-02 | 0.0 | 0.0 |

```
                              ***ROTOR DYNAMICS PROGRAM***

                    SEAL TESTER STABILITY WITHOUT LOADER, 4 BEARINGS

                              * BEARING PARAMETERS *

     BEARING NO.    1, AT STATION NUMBER     4,WITH  2 DEGREES OF FREEDOM.
    BEARING STIFFNESS

-0.8600E+06  0.0
 0.0        -0.8600E+06

     BEARING DAMPING

 0.0          0.0
 0.0          0.0


     BEARING NO.    2, AT STATION NUMBER     8,WITH  2 DEGREES OF FREEDOM.
    BEARING STIFFNESS

-0.8600E+06  0.0
 0.0        -0.8600E+06

     BEARING DAMPING

 0.0          0.0
 0.0          0.0
```

MODE SHAPE FOR FREQUENCY = 37336.0 RPM GROWTH FACTOR = 0.0



MODE SHAPE FOR FREQUENCY = 47325.7 RPM GROWTH FACTOR = -0.0



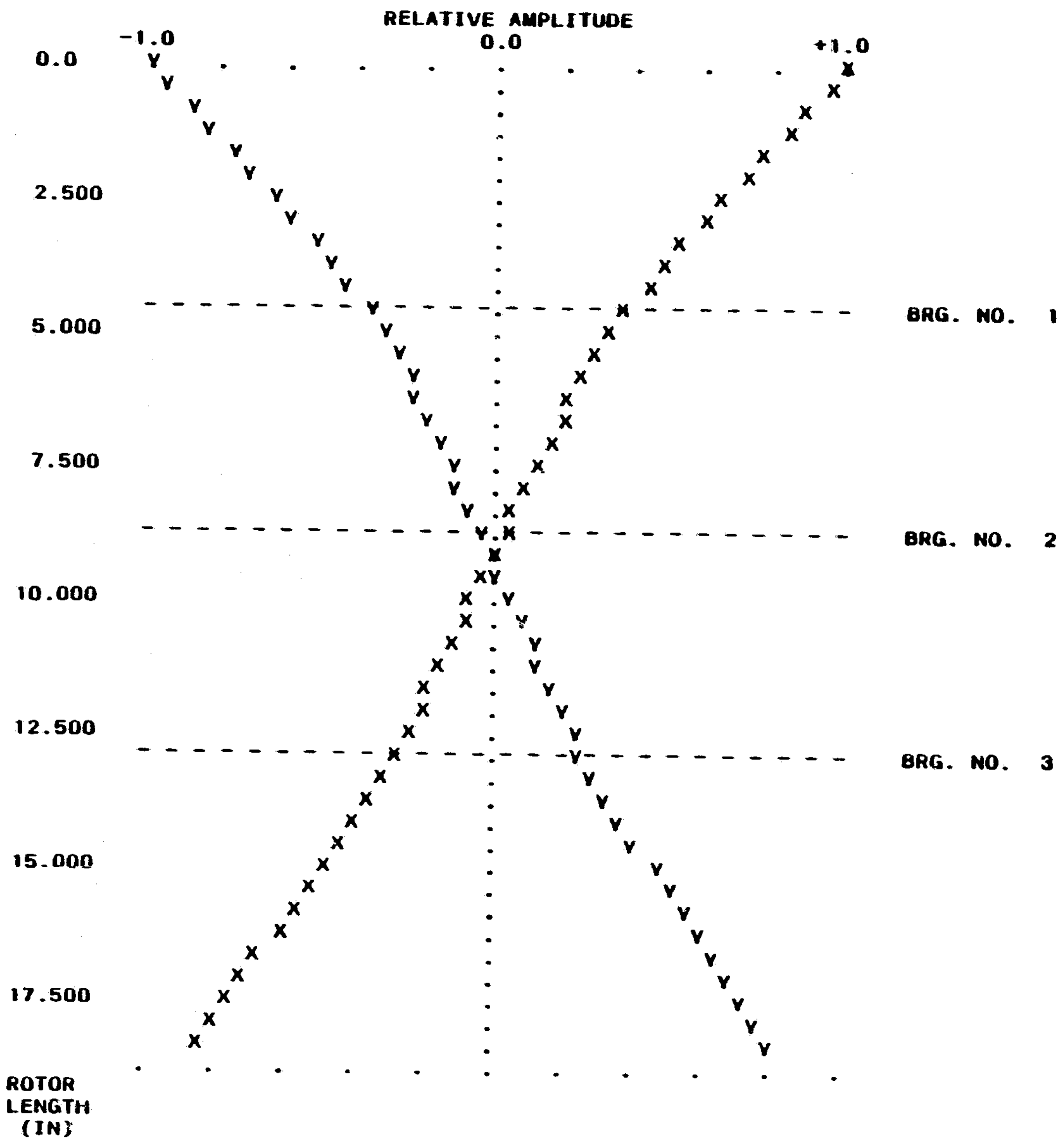
***ROTOR DYNAMICS PROGRAM***

SEAL TESTER STABILITY WITH LOADER, 4 BEARINGS,30000RPM, 1250PSI, .005IN

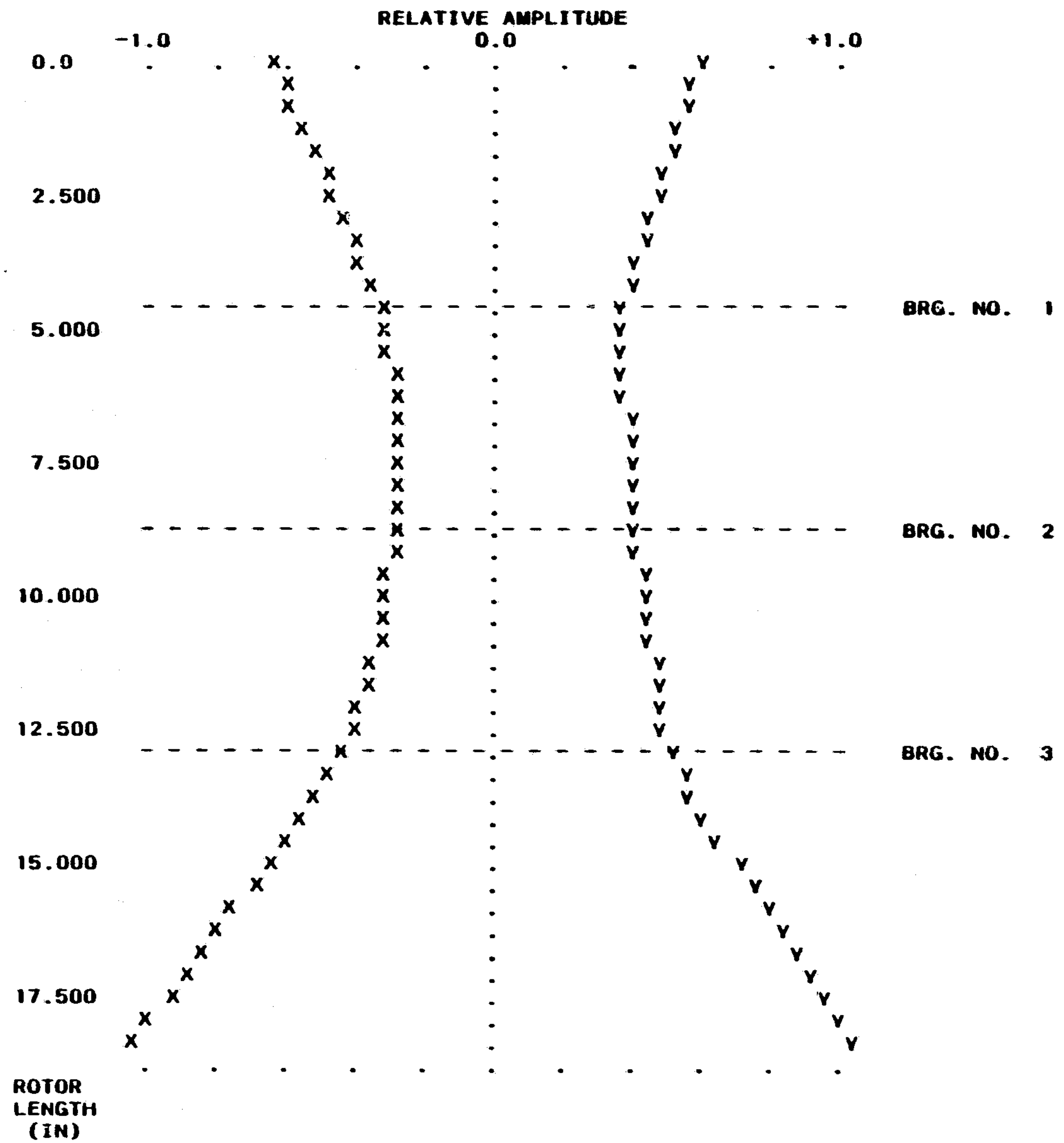
* BEARING PARAMETERS *

```
     BEARING NO.    1, AT STATION NUMBER     4,WITH  2 DEGREES OF FREEDOM.
   BEARING STIFFNESS

-0.8600E+06  0.0
 0.0        -0.8600E+06

     BEARING DAMPING

 0.0         0.0
 0.0         0.0


     BEARING NO.    2, AT STATION NUMBER     6,WITH  2 DEGREES OF FREEDOM.
   BEARING STIFFNESS

-0.2687E+06 -0.2360E+06
 0.1449E+06  0.6098E+05

     BEARING DAMPING

-0.1520E+03  0.0
 0.0        -0.9100E+02


     BEARING NO.    3, AT STATION NUMBER     8,WITH  2 DEGREES OF FREEDOM.
   BEARING STIFFNESS

-0.8600E+06  0.0
 0.0        -0.8600E+06

     BEARING DAMPING

 0.0         0.0
 0.0         0.0
```

MODE SHAPE FOR FREQUENCY = 37341.9 RPM GROWTH FACTOR = -0.9



MODE SHAPE FOR FREQUENCY = 48082.6 RPM GROWTH FACTOR = -257.8



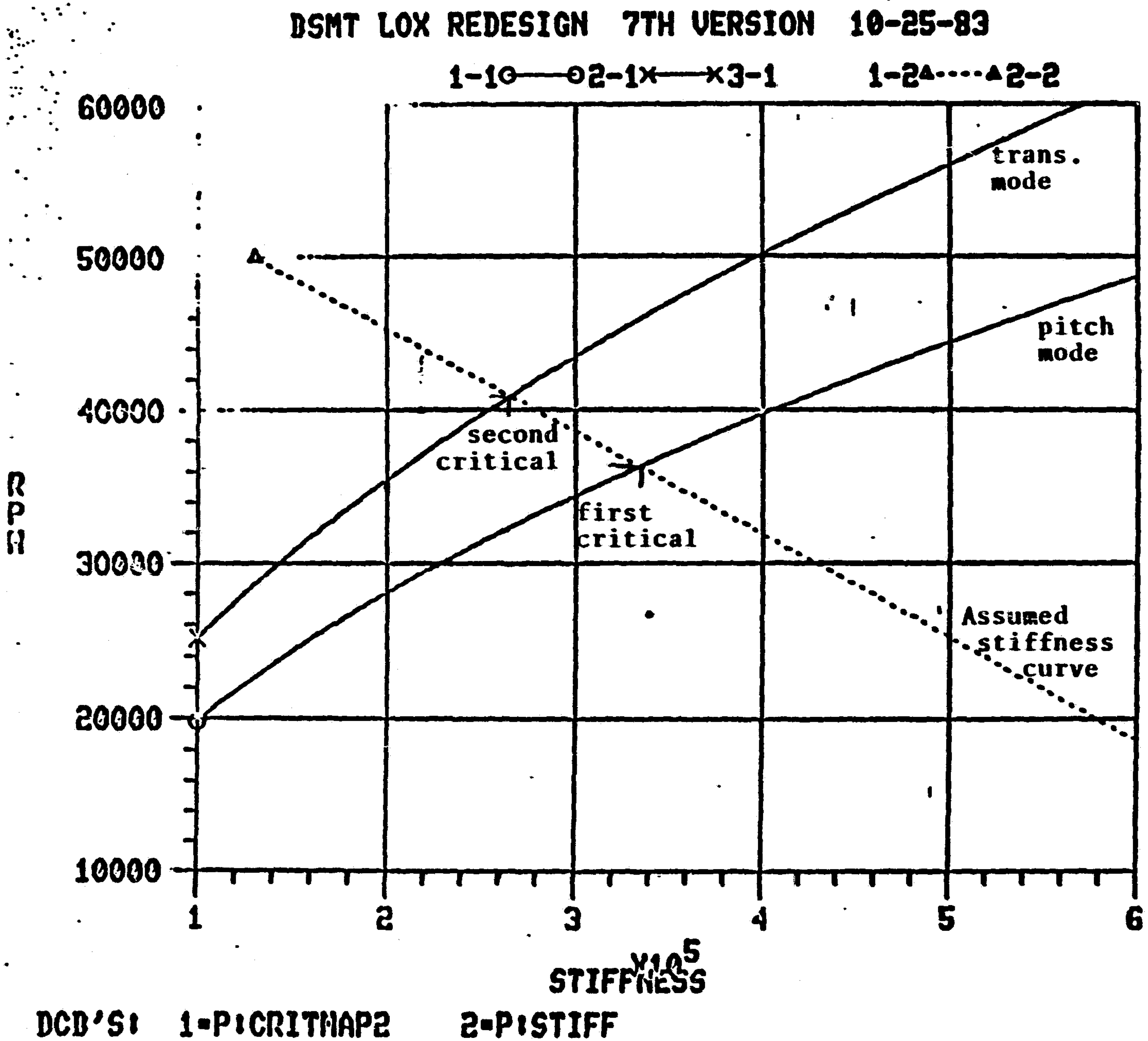

DSMT LOX REDESIGN 7TH VERSION 10-25-83
1-1 2-1 3-1 1-2 2-2
60000
50000
40000
30000
20000
10000
RPM
1
2
3
4
5
6
X10^5
STIFFNESS
trans. mode
pitch mode
second critical
first critical
Assumed stiffness curve
DCB'S: 1=P:CRITMAP2 2=P:STIFF

FIGURE 7:

# APPENDIX A

# DERIVATION OF G FACTORS

Definition of Child's friction factor

$$\nabla p = \frac{2\lambda_c}{h} \left(\frac{1}{2} \rho \bar{u}^2\right) \qquad . \quad (1)$$

Definition of 'G' factor for pressure driven turbulence

$$\bar{u} = \frac{h^2}{12\mu} \nabla p G \qquad (2)$$

Substitute Eq. (2) for $\bar{u}$ in equation (1) and solve for $\lambda_c$

$$\lambda_c = \frac{144}{G^2 R_e^*} \qquad (3)$$

where

$$R_e^* = \frac{h^3 \rho \nabla p}{\mu^2} \qquad (4)$$

"GBEAR" requires G as a function of $R_e^*$. $\lambda_c$ is given as a function of the "axial" Reynolds number $R_a$ defined as

$$R_a = \frac{2h\bar{u}\rho}{\mu} \qquad (5)$$

Substitute equation (2) for $\bar{u}$ in equation (5) and use equation (4) to put in terms of $R_e^*$ to obtain

$$R_a = \frac{1}{6} R_e^* G \left(R_e^*\right) \qquad (6)$$

For smooth surfaces such as the rotor we may calculate $\lambda_r$ from the Blasius equation

$$\lambda_r = n_r R^{m_r} \qquad (7)\ \text{(Rotor)}$$

where $n_r = 0.079$ and $m_r = -0.25$.

Childs has obtained measurements for two rough surfaces such as those making up the stator to obtain an empirical relationship of the form

$$\lambda_s = n_s R_a^{ms} \qquad (8)\ \text{(Stator)}$$

Childs further suggests averaging the friction factors for a smooth rotor and a rough stator as

$$\lambda_c = \frac{1}{2}(\lambda_r + \lambda_s)$$

This is very nearly equivalent to determine the 'G' factor for identical rotor materials, $G_r$, and for identical stator materials, $G_s$, and averaging them as suggested by Equation (3)

$$\frac{1}{G^2} = \frac{1}{2}\left(\frac{1}{G_r^{\,2}} + \frac{1}{G_s^{\,2}}\right) \qquad (9)$$

If we substitute Equ. (3) for $\lambda$ and Eq. (6) for $R_a'$ in equation (7) for the case of identical rotor materials we obtain

$$\frac{144}{G_r{}^2 R_e{}^*} = n_r \left(\frac{R_e{}^* G_r}{6}\right)^{mr}$$

which may be solved for $G_r$ as follows

$$G_r = \left[\frac{144}{n_r} \times 6^{m_r}\right]^{\frac{1}{2 + m_r}} R_e{}^{* - \frac{(1 + m_r)}{(2 + m_r)}} \qquad (10)$$

A similar relationship may be written for $G_s$

$$G_s = \left[\frac{144}{n_s} 6^{m_s}\right]^{\frac{1}{2 + ms}} R_e{}^{* \frac{-(1 + m_s)}{(2 + m_s)}} \qquad (11)$$

Equations (10) and (11) may be substituted for $G_r$ and $G_s$ in Equation (9) to obtain G.

APPENDIX B

WORK STATEMENT

# STATEMENT OF WORK

## Review of Lox Bearing and Seal Materials Tester (BSMT) Radial Load System

OBJECTIVES:

The objectives of this task are to: (1) review the NASA designed Lox BSMT Hydraulic Bearing Radial Load Concept as shown on drawing #30A85200-1 and details and (2) provide a written assessment concerning the feasibility of the design with recommendations for improvement as required to meet the operating constraints, fluids used, and the loading requirements as identified in BSMT-URD-83-3, dated March 25, 1983. Special attention to safety of operation in lox, ability to achieve desired loading, flow requirements, instrumentation recommendations and an assessment of the load predictability is desired.

REQUIREMENTS:

The concept is described in MSFC drawing 30A85200-1 and the requirements are identified in BSMT-URD-82-3.

With these documents as controls and in concert with the MSFC COR, the contractor is expected to make a technical assessment of the MSFC design in terms of the objectives as outlined above. Specifically, a new concept effort is not expected, but recommendations as to the feasibility of the existing concept and/or any feasible modifications to improve the function and safety of the existing concept are expected. Rationale and technical analysis to back up recommendations should be provided.

APPROACH:

In performing this task the contractor shall give special attention to defining operational hazards in lox, the ability to accurately determine loading on the shaft and/or bearings and to what magnitude of accuracy, defining the recommended load application technique if different than that proposed. The contractor shall provide the necessary manpower, expertise and resources to execute the following tasks.

<u>Analysis</u> - Sufficient analysis to verify the capability of the design to perform the loading tasks required when functioning as a fluid bearing, including a definition of all expected forces operational on the shaft. Fluid supply is limited to 2000 psi. If greater supply pressure is required it should be identified.

<u>Sketches</u> - Sketches of proposed modifications are to be furnished. They <u>need not</u> be scaled, but required dimensions should be identified. Hand drawings are acceptable.

<u>Safety</u> - The current design provides a 5 $\pm 1$ mil clearance at operating temperature in lox between the bearing and shaft. In addition to the nominal clearance it is expected that an additional 1 to 2 mils will be generated by shaft flexure and bearing movement. However, any obvious safety hazard from past utilization of hydrostatic bearing applications should be reviewed in the context of this application and an opinion formulated and provided.

INITIAL REVIEWS:

It is contemplated that the contractor will meet initially with MSFC to review the background and obtain data necessary to fulfill the objectives. Not more than two personnel should attend such a review.

FINAL REVIEW:

At the conclusion of the design review, one contractor representative should present all findings to the COR and other interested MSFC personnel.

DOCUMENTATION REQUIREMENTS:

One formal report describing the effort above and providing the technical analysis is expected. It should be provided with ten copies and one reproducible copy.

A handout of the final review charts and/or viewgraphs with ten copies should be provided.